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THE HIGH-SPEED
NTERNAL-COMBUSTION
ENGINE

BY

HARRY R. RICARDO, F.R.S.

Gold Medallist of the North-East Coast Institution of Engineers and Shipbuilders
Silver Medallist of the Royal Aeronautical Society
Institution Medallist, Inst. of Automobile Engineers
Crompton Medallist, Inst. of Automobile Engineers

REVISED BY

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PREFACE TO THIRD EDITION

The present volume is the third edition of Vol. II of *The Internal-Combustion Engine*, first published in 1923. Owing to the European war and the strenuous months preceding it, I have had little or no time to undertake this work of revision, and have therefore entrusted it to my very able assistant, Mr. H. S. Glyde, who has worked with me during the last seventeen years. During the whole of this time he has been in direct personal touch with the researches carried out in my firm's laboratory at Shoreham.

Mr. Glyde is himself the author of a number of technical papers dealing with various aspects of the Internal-combustion Engine, and has devoted himself more particularly to the difficult task of checking over, analysing and co-ordinating the results of research on this subject. He is, I think, peculiarly well fitted to undertake this work.

Mr. Glyde has gone carefully through the 1931 edition, has checked over and added to the original matter in the light of more recent findings, and has rewritten entirely the chapter relating to Diesel engines. The projected chapter on aero-engines has been omitted meantime.

H. R. R.

FOREWORD

When invited to revise this book, the writer felt some diffidence in undertaking a task which inevitably involved making alterations and additions to a work so clearly bearing the impress of the authority and personality of its author. His object has been first to retain as much as possible of the original, the greater part of which dealt with principles of engine design and fuel behaviour which have not changed; secondly, to supply new subject-matter regarding the high-speed Diesel engine. (The projected chapter on aero-engines has been omitted meantime.) At the same time certain parts which have rather lost their topical interest have been removed. If the work, after this treatment, provides the same interest to-day as its earlier editions in their day, the writer is well content.

Grateful thanks are due to Mr. H. R. Ricardo for his kindly encouragement and guidance, and for something less easily defined, the infection of his unfailing enthusiasm and buoyancy of spirit.

The writer is grateful also to Messrs. G. L. Ensor and C. O. B. Beale for much helpful criticism and for reading the proofs, and to Messrs. M. Mead, C. A. Beard and J. H. Pigneguy for valuable information and assistance. He acknowledges with thanks the help given by Mr. Maurice Platt and Messrs. Vauxhall Motors, who supplied data and drawings. To Mr. E. J. F. Sumner, whose skilful preparation of the typescript has gone far beyond the mere typing of it, the writer also gives his thanks.

H. S. G.

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THE HIGH-SPEED INTERNAL-COMBUSTION ENGINE

INTRODUCTION

Until the war of 1914–18, most of the scientific talent in this country which had interested itself in the development of the internal-combustion engine was devoted to research upon, and development of, the relatively heavy slow-speed stationary type, a type which, it now appears, has but a limited scope, for in the really large powers it can hardly compete with the modern steam turbine, while in the smaller powers, its field is being narrowed daily by the gradual spread of electricity.

Very soon after hostilities began, it became apparent that the light mobile high-speed type of internal-combustion engine applied to transport, aircraft, and later to tanks, was destined to play a very important, if not decisive, part in the conduct of the war. Every effort was then made to concentrate all the available scientific talent on the development of the high-speed engine. Independent scientists and investigators, and such national institutions as the Royal Aircraft Establishment and the National Physical Laboratory, were requested to turn their attention to this subject, and every facility was lavished on them. They were invited to co-operate with the manufacturers and were asked to make a careful theoretical study of both the mechanical and thermodynamic problems involved, and to recommend how and in what direction the general efficiency of these light engines might be maintained and improved. The campaign of intensive research which resulted from this sudden influx of scientific talent, accompanied by almost inexhaustible funds for research, has resulted in the production of light high-speed engines which, besides giving what twenty years ago would have been considered an almost incredible power output in relation to their size and weight, can show as high an efficiency as that of the largest slow-speed type. What is perhaps more important still, the basic principles, both mechanical and thermodynamic, upon which the performance of such an engine depends, have been investigated in so complete and comprehensive a manner that the performance of any engine can now be gauged *with accuracy from a study of the design alone; or conversely, an engine can be designed to fulfil any specific requirement as to power output or efficiency, with the same precision as in the case of a steam-engine.*

That the internal-combustion engine has found its ultimate sphere in the light mobile high-speed type is now evidenced by the fact that, whereas in the years immediately before the war of 1914–18 the annual output in horse-power of both the light and heavy type in this country was about equal, to-day the aggregate annual power output of the light high-speed type is many many times that of all other types.

To-day far more is known about both the possibilities and the limitations of the high-speed internal-combustion engine than was the case twenty-five years ago, and it seems fairly evident that its role lies in the propulsion of all forms of transport, where its light weight and low fuel consumption render it supremely valuable. It is in the author's opinion extremely doubtful whether it will ever attain to more than its present very uncertain footing for stationary purposes, where neither its own light weight nor that of the fuel it consumes can be of much assistance to it in the struggle for existence.

Already, and in an incredibly short space of time, the internal-combustion engine has gained practically undisputed sway over all forms of road transport, and in doing so has developed and even almost revolutionized this previously decaying system. In a few short years it has both opened up the possibility of aerial transport and made it a powerful factor in war. It is being widely applied also to rail transport, more particularly in countries where long distances have to be traversed and where fuel and water are scarce. It has already ousted the steam-engine from the smaller classes of shipping and is extending, though more slowly, to the larger vessels. The reason for its slower progress here lies in the fact that the steam-turbine shows to particular advantage as a marine engine, because in this field it can always get that upon which its efficiency so largely depends, namely, an unlimited supply of cold water. Also the large steamship, alone of all forms of transport, requires a very high-powered installation, and it is in units of high power output that steam retains its supremacy.

With the exception of those using heavy oil, all light mobile engines consume volatile liquid fuels.

In the earlier days the only two fuels available in bulk were petrol—a generic term covering any low-boiling distillate from crude petroleum—and benzol, a distillate from coal-tar, consisting of benzene, with a small percentage of toluene. The demands of the light high-speed engine soon reached such proportions that it became clear that other ways and means must be found to meet them. The necessary production of motor spirit by straight distillation of the lighter fractions would have involved a vast output of crude oil from the wells of the world, and a colossal wastage of the heavier distillates. This situation, critical because of the serious depletion of the oil reserves, has been dealt with by the development of cracking and polymerization processes, enabling fuels of excellent anti-knock properties to be produced from almost any of the heavier fractions of crude oil. In this way, crudes previously regarded as unrefinable and useless have yielded valuable products.

The extent to which the world's supply of crude oil has been conserved is shown by the fact that more high-grade spirit is produced by cracking to-day than by the normal straight distillation process. Even so, the demand is so great, and the inroad into natural resources so deep, that every effort should be made to produce fuels from alternative sources. Civilization is now so deeply committed to the use of internal-combustion engines for all road transport and for many other purposes, that it is well to be prepared to deal with a critical situation in the fuel supply should it arise. Drilling operations in Great Britain have so far not yielded a single producing well.

A little progress has been made in the production of fuel from coal by the high-temperature carbonization process, which yields a spirit consisting mainly of benzol.

The production of alcohol from vegetable sources would involve no drain on the world's storage and could, in tropical countries at all events, ultimately produce fuel in sufficient quantity for the world's needs.

By the use of a fuel derived from vegetation, mankind is adapting the sun's heat to the development of motive power, as it becomes available from day to day; by using mineral fuels, he is consuming a legacy—and a limited legacy at that—of heat stored away many thousands of years ago. In the one case he is, as it were, living within his income, in the other he is squandering his capital.

The mobile internal-combustion engine is now no longer a luxury; it has become one of the prime necessities of peaceful civilization and the prime necessity in time of war; therefore, the assurance of its fuel supply should be considered a matter of national importance. It is perfectly well known that alcohol is an excellent fuel, and there is little doubt but that sufficient supplies could be produced within the tropical regions of the British Empire, yet little or nothing is being done to encourage its development.

Recent years have been marked by the development of the high-speed Diesel engine—a power unit using relatively heavy and high-boiling fuels such as cannot be vaporized outside the cylinder. This has been the most outstanding achievement of internal-combustion engine research since the war of 1914–18. In road and rail transport, and to a much less degree in the air, the high-speed Diesel engine is doing work previously done by the petrol engine. A new fuel industry has been built up to supply its needs, and methods of measurement of fuel quality have been extended to include the standardization of fuel oils.

In the author's opinion it is unlikely that any crisis in the fuel situation will have an adverse effect on the development of the internal-combustion engine, for the simple reason that it has become a necessity, but it will probably have the effect of increasing the cost of transport and with it the cost of living generally. It will also, of course, have the effect of forcing designers of engines to concentrate more attention on the attainment of high efficiency and fuel economy, which is all to the good.

CHAPTER 1

VOLATILE LIQUID FUEL FOR INTERNAL-COMBUSTION ENGINES

Petrol, as is well known, is a distillate from crude petroleum; it consists of a heterogeneous mixture of all those hydrocarbon fractions which boil between the limits of 140° F. and about 400° F.

These fractions belong to three different series:

	General Formula
The Paraffins	C_nH_{2n+2}
The Naphthenes	C_nH_{2n}
The Aromatics	C_nH_{2n-6}

In addition to these three leading groups there are also present a small proportion of members of the olefine series, though the proportion found in " natural " as opposed to " cracked " petrols is usually so small as to be almost insignificant.

The individual members of the paraffin series present in petrol are:

Fuel.	Formula.	Boiling Point 0° F.	Specific Gravity at 60° F.
Hexane	C_6H_{14}	156	0·663
Heptane	C_7H_{16}	209	0·691
Octane	C_8H_{18}	258	0·709
Nonane	C_9H_{20}	302	0·723
Decane	$C_{10}H_{22}$	343	0·735
Undecane	$C_{11}H_{24}$	383	0·746

Those of the naphthene series are:

Fuel.	Formula.	Boiling Point 0° F.	Specific Gravity at 60° F.
Cyclohexane	C_6H_{12}	178	0·780
Hexahydrotoluene	C_7H_{14}	212	0·770
Hexahydroxylene	C_8H_{16}	246·2	0·756

and those of the aromatic series :

Fuel.	Formula.	Boiling Point 0° F.	Specific Gravity at 60° F.
Benzene	C_6H_6	176	0·884
Toluene	C_7H_8	230	0·870
Xylene	C_8H_{10}	284	0·862

Although, in most examples of commercial petrol, the members of the paraffin series predominate, yet this is by no means always the case, and it is the exception rather than the rule, for the paraffin content of a petrol to exceed 60 per cent of the whole. Generally speaking, paraffins are commonest in the Western oilfields, naphthenes in the near Eastern, and aromatics in the far Eastern oilfields. This, however, is only a rough generalization, for there are many exceptions.

The following table gives the analysis of seven typical samples of petrol drawn from widely different parts of the world and illustrates how greatly the composition may vary. In this table the presence of small traces of other complex substances such as thiophene, &c., which play no perceptible part in the behaviour of the fuel, has been ignored.

Sample.	Approximate Composition by Weight per cent.			Specific Gravity.
	Paraffins.	Naphthenes.	Aromatics.	
A	26·0	35·0	39·0	0·782
B	62·0	23·0	15·0	0·723
C	61·0	30·5	8·5	0·727
D	38·0	47·0	15·0	0·760
E	68·0	20·0	12·0	0·719
F	80·0	15·2	4·8	0·704
H	10·0	85·0	5·0	0·767
Average	49·3	36·5	14·2	0·740

It is evident from the above that the specific gravity as a measure either of the composition or of the volatility of a fuel is quite meaningless. If, as is sometimes erroneously supposed, petrol consisted entirely of members of the paraffin series, then, since the gravity, the molecular weight, and the boiling point of the members of this series rise together, the gravity would be a measure of the volatility. The presence, however, of even a very small proportion of aromatics whose specific gravity ranges from 0·860 to about 0·885 will, of course, entirely upset any deductions which could be drawn from considerations of specific gravity alone. As a specific illustration it may be mentioned that of the fuels tabulated above, sample B sp. gr. 0·723, a special low-boiling aircraft spirit prepared for the Cross-Atlantic flight of 1919, is by far the most volatile, yet its gravity is not the lowest by any means. The best fuels, i.e. those from which the highest power and efficiency could be obtained, were samples A, D, and H, which have the highest specific gravity of all, but which are rich in aromatics or naphthenes or both, while the worst without question is sample E sp. gr. 0·719.

It will be shown later that, of the three leading groups, the presence of the aromatics is the most of all to be desired from every point of view, that of the naphthenes next, while the paraffins are highly objectionable and the smaller the proportion present the better.

Though the phenomenon of detonation will be discussed later it may be stated at this stage that it is by far the most important factor in determining the quality of a fuel, and it is one which depends primarily upon its chemical composition. The paraffin series are the worst of all from this point of view, and they become progressively worse as their molecular weight and gravity increase; thus, for example, hexane is much better than heptane, and so on. The naphthenes are very much better, while the aromatics are the best of all as regards detonation.

Commercial benzol is a coal-tar distillate consisting primarily of pure benzene C_6H_6, with a little toluene and a trace of xylene. These are all aromatics. Its specific gravity ranges from 0·875 to 0·882, depending on the proportion of toluene present.

This fuel has many advantages over petrol, but fully to realize these, it is necessary to work with a much higher compression ratio.

The available members of the alcohol group consist of methyl, ethyl, and butyl alcohol. These are not true hydrocarbons, since each contains oxygen in the molecule. In consequence of this and of their higher latent heat of evaporation, the behaviour of these fuels is somewhat different from that of the true hydrocarbons. So far as their tendency to detonate is concerned they are even better than the aromatics, though when used under a very high compression methyl alcohol in particular is liable to pre-ignite without warning. Owing to their high latent heat and low flame temperature the whole temperature of the cycle is lower, while owing again to their high latent heat of evaporation and therefore to the reduced suction temperature, the volumetric efficiency of an engine using alcohol is considerably higher than when using petrol or benzol. The result of the lower flame temperature is that the engine operates at a higher thermal efficiency, while the increase in volumetric efficiency much more than balances their lower internal energy, so that the maximum power output on alcohol is considerably greater, while the heat flow to the engine cylinder is lower than on petrol or benzol.

The properties of a fuel which determine its value for use in an internal-combustion engine are:

(i.) Tendency to detonate.
(ii.) Latent heat.
(iii.) Volatility.
(iv.) Calorific value of the fuel.
(v.) Heat value of the mixture.

All volatile liquid fuels when vaporized and mixed with air in the proportion required to give complete combustion have, within very close limits, the same heat value per standard cubic inch of mixture, hence they all give the same power and the same thermal efficiency when used under the same conditions. It is only by varying the compression ratio or by altering the degree of vaporization in the carburettor or induction pipe that any variation in power output or efficiency can be obtained.

Tendency of Fuels to detonate.—The phenomena of detonation as apart from its relation to the nature of the fuel will be dealt with later, but for the present it is sufficient to state that the limit to which the compression ratio can be raised, and therefore the limit of power output and efficiency, is governed by the conditions which control detonation and pre-ignition.

With all known petroleum spirits, detonation precedes and subsequently produces pre-ignition, but in the case of certain fuels such as ether, carbon disulphide (at a low compression ratio), also the pure aromatics and alcohol (at a high compression ratio), pre-ignition is liable to occur without preliminary detonation.

Of the constituents of petrol the paraffins are by far the worst offenders as regards detonation, while the aromatics are the best. It is found also that all mixtures of these bodies obey the ordinary proportional laws and that there is practically a straight line relation between the mixture proportion of two or more hydrocarbons and the compression ratio at which detonation occurs.

In fig. 1 is shown as a full line the observed relation between the compression ratio and the point at which detonation occurs when to pure heptane varying proportions of benzene (benzol) are added. Owing to the influence of combustion-chamber design, and to other factors which will be considered later, it is not possible to lay down a hard and fast relation between the fuel and the highest compression ratio at which it may be used in any type of engine, but it is possible in the light of present knowledge to give relative values, though there is some difficulty in selecting substances to be taken as standards. In the investigations which the authors' firm carried

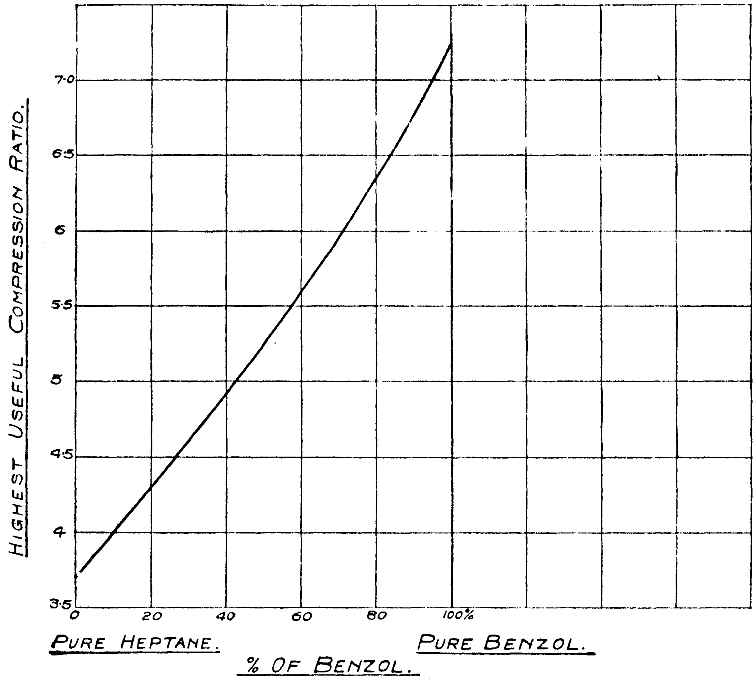

Fig. 1.—Curve showing Compression Ratio at which Detonation occurs

out for the Asiatic Petroleum Co. in 1919, a sample of petrol, consisting mainly of paraffins from which nearly all the aromatics had been removed by sulphonation, was taken as zero, and at the other end of the scale pure toluene was chosen; the relative tendency of different fuels to detonate was then expressed in terms of their " toluene value," i.e. the equivalent proportion of toluene which it would be necessary to mix with the standard aromatic free petrol in order to give it the same tendency to detonate as that of the sample under examination. Later investigation showed that the standard " aromatic free " petrol, which contained about 35 per cent of naphthenes and the lighter members of the paraffin series, was by no means the worst offender as regards detonation, and that in fact several samples of commercial petrol were actually considerably worse. Also it was found that toluene was not so effective in resisting detonation as ethyl alcohol. Since, however, the expression " toluene value " had become rather widely used, it was decided to retain the term.

Table I gives the toluene values and the highest useful compression for various fuels. The highest useful compression ratio may be defined as the highest ratio at which a particularly efficient engine used for the purpose of investigating the behaviour of fuels could be operated without detonation at any mixture strength or with any ignition timing, with a standard amount of preheating to the carburettor, and at a speed of 1500 R.P.M.

It was therefore purely a relative term; that is to say, its absolute value applied only to one particular type of engine operated under one given set of conditions, but its relative value was applicable to any type of engine and under any conditions, as will be shown later.

In this connection reference may be made to the common belief that the rate of burning of the fuel, though one of the factors controlling detonation, forms a limit to the speed at which an engine can run.

The normal rate of burning (as distinct from the detonation rate) of any stagnant fuel/air mixture is so low as to be practically useless so far as any internal-combustion engine is concerned. We must look, therefore, entirely to turbulence or the mechanical distribution of the flame to spread combustion throughout the whole mass of the working fluid, and, since this is the case, it follows that the normal rate of burning of any fuel is practically without influence on the speed at which an engine will run.

It has been found that a fuel with a low normal rate of burning,

TABLE I

Fuel.	Highest Useful Compression Ratio in variable Compression Engine. $\frac{\text{Total Volume}}{\text{Clearance Volume}}$.	Toluene Value. Toluene = 100 %, Aromatic Free Petrol = 0 %.
Aromatic Free Petrol	4·85	0
" A " Petrol	6·0	38·0
" B " ,, ...	5·7	28·0
" C " ,, ...	5·25	13·5
" D " ,, ...	5·35	16·5
" E " ,, ...	4·7	- 5·0
" F " ,, ...	5·05	6·5
" G " ,, ...	4·55	- 10·0
" H " ,, ...	5·9	35·0
" I " ,, ...	4·3	- 20·0
Heavy Fuels		
Heavy Aromatics	6·5	55·0
Kerosene	4·2	- 22·0
Paraffin Series		
Pentane (Normal)	5·85	33·0
Hexane (80 % pure)...	5·1	8·0
Heptane (97 % pure)	3·75	- 37·0
Aromatic Series		
Benzene (pure)	6·9 †	67·0
Toluene (99 % pure)...	>7·0	100·0
Xylene (91 % pure)	>7·0	85·0
Naphthene Series		
Cyclohexane (93 % pure)	5·9 †	35·0
Hexahydrotoluene (80 %)	5·8	31·5
Hexahydroxylene (60 %)	4·9	1·5
Olefines		
Cracked Spirit (53 % unsat.) ...	5·55	23·5
Alcohol Group, &c.		
Ethyl Alcohol (98 %)	>7·5	>88·0
,, ,, (95 vol. %)	>7·5	>88·0
Methyl Alcohol (Wood Naphtha) ...	5·2 †	..
Methylated Spirits	6·5 †	..
Butyl Alcohol (Coml.)	7·3	80·0
Ether (50 % in petrol)	3·9	(- 32·0)
Carbon Disulph. (50 %)	5·15 †	(9·0)

NOTE.—This sign (†) indicates that pre-ignition occurred before audible detonation.

such, for example, as ethyl alcohol, will operate just as efficiently in a high-speed, low-compression, engine as will hexane or petrol, and that the relative efficiency and power obtained is exactly the same throughout the whole speed range. The normal rate of burning of a fuel has, therefore, no connection with the speed at which an engine may be run.

In fact, since a slow-burning fuel is generally less prone to detonation, it is much more desirable than one which burns more rapidly. It is only when using excessively weak and therefore slow-burning mixtures, or when the normal turbulence within the cylinder is unduly low, that the influence of a naturally slow-burning fuel can be felt at all, and, even so, this can always be counteracted by a small advance of the ignition timing. Under all normal working conditions the difference in respect to engine speed between a normally slow or rapid burning volatile liquid fuel is quite imperceptible.

Latent Heat.—The influence of the latent heat of evaporation of the fuel is a very important factor, but one which is usually ignored. This factor, coupled with the mean volatility, determines the density of the charge taken into the cylinder. It is, of course, clear that the weight of charge taken into the cylinder will, in any given case, be inversely proportional to its absolute temperature at the moment when the inlet valve closes. There is definite evidence from experimental results that, with the exception of alcohol and the other members of its group, all fuels boiling below about 400° F. are completely evaporated before the commencement of the compression stroke by contact with the hot walls and by admixture with the highly heated residual exhaust products in the cylinder; excepting only a very small proportion which may enter the cylinder in coarse drops, and so not only escape evaporation, but, even to a large extent, combustion also. This proportion is, however, quite insignificant, and has no influence, in so far as power output is concerned.

The absolute temperature at the commencement of the compression stroke is dependent upon (*a*) the amount of external heating applied, and (*b*) the latent heat of evaporation. It is largely independent of the temperature of the mixture during its entry to the cylinder. In fact, the final absolute temperature, and, therefore, the weight of the charge taken into the cylinder, are dependent upon the quantity and latent heat of the fuel, and upon the amount of heat added to it, external to the cylinder.

For example, a highly volatile fuel entering the cylinder at 40° F., and a fuel of low-vapour tension entering at 80° F., will both have the same final absolute temperature at the commencement of compression, if the latent heat of both is the same, and if both receive the same amount of pre-heating. In the former case most of the evaporation has taken place outside the cylinder, and the added heat has been absorbed by the latent heat of evaporation; in the latter case, little or no evaporation has taken place outside the cylinder, and the added heat has therefore raised the temperature of the air and of the, still liquid, fuel. In both cases contact and admixture with the highly heated exhaust products in the cylinder will complete evaporation, and in both cases the final temperature will be the same, hence the weight of working fluid (which is inversely proportional to the absolute temperature at the end of the suction stroke), and therefore the power output, will be the same in both cases.

From the above considerations it will be seen that, with any given amount of pre-heating (provided it is not excessive), the volumetric efficiency, and therefore the power output, will increase with the latent heat of the fuel.

TABLE II

Name of Fuel.	Latent Heat of Evaporation, B.Th.U.s per lb.	Total Energy liberated by Combustion. Ft. lb. per standard cub. in.	Relative Power, Output allowing for Increase in Density due to Evaporation. Octane = 100.
Paraffin Series			
Hexane...	156	48·33	100·2
Heptane	133	48·64	100·1
Octane	128	48·73	100·0
Nonane...	..	48·78	..
Decane	108	48·82	99·4
Aromatic Series			
Benzene	172	47·51	100·1
Toluene...	151	47·98	100·0
Xylene	145	48·26	100·6
Naphthene Series			
Cyclohexane	156	48·11	100·0
Hexahydrotoluene ...	138	48·32	99·8
Hexahydroxylene ...	133	48·49	99·8

Some calculations illustrating this point more fully will be found in Chapter II.

The above conclusions may be summed up as follows:—

(1) The power output is inversely proportional to the absolute temperature of the working fluid at the end of the suction stroke—since it is the temperature at this point which controls the weight of charge, and therefore the volumetric efficiency.

(2) Other things being equal, the final suction temperature is controlled by the amount of external heating, on the one hand, and the latent heat of the fuel, on the other; it is nearly independent of the temperature of the entering charge and of its volatility.

Apart from the alcohol group, the variation in latent heat is not very large, and does not exercise any important influence. It is interesting to note, however, that in cases where the total internal energy is lower, the latent heat is generally slightly higher; consequently a slightly greater weight of charge is taken into the cylinder, sufficient in most cases to compensate for the lower internal energy, and thus bring the actual power output to substantially the same in all cases.

This point is well illustrated by the instance of benzene, as will be seen later, or by reference to Table II. The energy liberated by the combustion of a cubic inch (at standard temperature and pressure) of benzene-air mixture is appreciably lower than that of the hydrocarbons forming the greater proportion of petrols. On the other hand, the latent heat of benzene is considerably greater, and as a result the power output obtainable under similar conditions from benzene is the same as that from petrol to within less than one half of 1 per cent.

The following Table III gives the latent heat of evaporation of a number of hydrocarbons and other substances enumerated in the previous tables. The air-to-fuel ratio by weight, also the drop in temperature of the mixture due to evaporation of the liquid, are shown for each fuel. The calculations are made for mixtures giving complete combustion, but without excess of air.

In the case of alcohol, owing to the very much higher latent heat and to the fact that the proportion of fuel to air is also much greater, the latent heat of evaporation plays a supremely important part, and results in a really marked increase in power as compared with other fuels, although the total internal energy of unit mass of mixture is lower than that of either petrol or benzol. Moreover, there is introduced a feature which is not observed to any marked

TABLE III

Name of Fuel.	Latent Heat of Evaporation, B.Th.U.s per lb.	Air-to-fuel Ratio (by Weight) for just complete Combustion.	Fall in Temperature of Mixture due to Latent Heat of Evaporation, ° F.
Paraffin Series			
Hexane	156	15·2	37·8
Heptane	133	15·1	32·4
Octane	128	15·05	29·0
Nonane	..	15·0	..
Decane	108	15·0	20·1
Aromatic Series			
Benzene	172	13·2	46·8
Toluene	151	13·4	40·5
Xylene	145	13·6	38·7
Naphthene Series			
Cyclohexane	156	14·7	38·7
Hexahydrotoluene ...	138	14·7	34·2
Hexahydroxylene ...	133	14·7	32·4
Olefine Series			
Heptylene	167 (app.)	14·7	41·4
Decylene	..	14·7	..
Alcoholic Group			
Ethyl Alcohol	397	8·95	148·8
Methyl Alcohol ...	512	6·44	252·0
Miscellaneous			
Ether	158	11·14	49·5
Carbon Disulph. ...	153	9·35	55·8
Acetylene	Gas	13·2	..
Carbon Monox. ...	Gas	2·45	..
Hydrogen	Gas	34·3	..

The last column is calculated on the assumption that the specific heat of the fuel vapour is constant for all at 0·5.

extent with other fuels—namely, that the power output increases very considerably when an over-rich mixture is used, because more fuel is then evaporated, the temperature of the charge is lowered, and the gain in weight of charge considerably more than outweighs the loss due to the greater specific heat of the products of combustion.

In fig. 2 are shown the actual measured volumetric efficiency as found in the author's fuel research engine when using petrol and ethyl alcohol under precisely similar conditions as to temperature, &c., and at a compression ratio of 5 : 1. In both cases a careful series of measurements was made at mixture strengths ranging from 20 per cent weak to 25 per cent over-rich.

In the case of fuels whose volatility is very low, such as kerosene, butyl alcohol, &c., advantage cannot be taken of the latent heat of evaporation, because it then becomes necessary to add an excessive amount of heat before entry to the cylinder, in order to prevent condensation in the induction system. For this reason alone the power output obtainable from kerosene is actually some 15 per cent lower than from petrol or other volatile hydrocarbons at the same compression ratio.

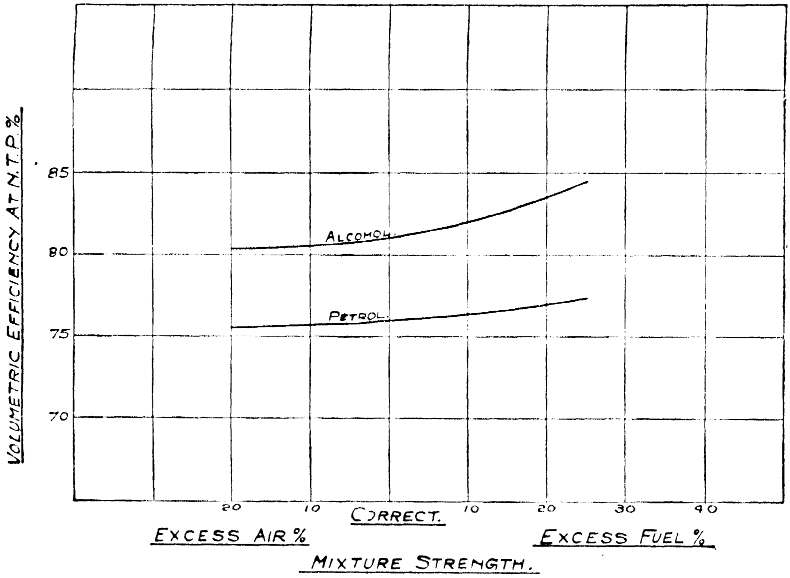

Fig. 2.—Observed Volumetric Efficiency on Petrol and Alcohol at different Mixture Strengths

Volatility.—The mean volatility of a fuel is of importance since this determines the amount of pre-heating required to give reasonably uniform distribution. The amount of pre-heating governs, in its turn, the use which may be made of the latent heat of the liquid fuel.

In single-cylinder engines volatility is, between wide limits, of comparatively little consequence since the exposed surface of the induction pipe is relatively small, but as the number of cylinders is

TABLE IV

Fuel.	Rise or Fall of Temp. in Induction Pipe (indicating app. Mean Volatility). Heat 65 $\frac{\text{B.Th.U.}}{\text{Min.}}$ Deg. F.	Fall in Temperature (calculated) of $\frac{\text{Air}}{\text{Fuel}}$ Mixture due to Latent Heat of Evaporation. Deg. F.
Aromatic Free Petrol	+ 18	32·4
" A " Petrol	+ 19·8	36·0
" B " ,, ...	+ 5·4	34·2
" C " ,, ...	+ 12·6	33·3
" D " ,, ...	+ 19·8	33·0
" E " ,, ...	+ 18·9	32·8
" F " ,, ...	+ 9·5	32·8
" G " ,, ...	+ 32·4	...
" H " ,, ...	+ 21·6	36·0
" I " ,, ...	+ 25·2	...
Heavy Fuels		
Heavy Aromatics	+ 50·5	35·7
Kerosene	+ 56	26·1
Paraffin Series		
Pentane (Normal)	...	37·8
Hexane (80 % pure)...	0	37·8
Heptane (97 % pure)	+ 10	32·4
Aromatic Series		
Benzene (pure)	– 13·1	46·8
Toluene (99 % pure)...	+ 14·4	40·5
Xylene (91 % pure)	+ 32·4	38·7
Naphthene Series		
Cyclohexane (93 % pure)	– 5·4	38·7
Hexahydrotoluene (80 %)	+ 5·4	34·2
Hexahydroxylene (60 %)	+ 24·3	32·4
Olefines		
Cracked Spirit (53 % unsat.) ...	+ 19	37·5
Alcohol Group, &c.		
Ethyl Alcohol (98 %)	+ 2·7	153·0
,, ,, (95 vol. %)	– 3·6	176·0
Methyl Alcohol (Wood Naphtha) ...	– 15·3	252·0
Methylated Spirits	– 1·8	198·0
Butyl Alcohol (Coml.)	+ 18	...
Ether (50 % in petrol)	+ 1·8	39·6
Carbon Disulph. (50 %)	– 12·6	48·6

multiplied and the length and surface of the induction system increased, so does the importance of volatility increase. A rough approximation (it is no more) of the relative volatility of different fuels can be obtained by measuring the rise or fall of temperature in the induction pipe of an engine, when a known weight of fuel and air are passing into it and when the amount of heat supplied to the carburettor can be recorded accurately.

The figures in Table IV (p. 17) give some clue as to the relative volatility of the different fuels—all were tested under exactly the same conditions as regards speed, temperature, heat input, &c. The temperatures measured in the induction pipe and recorded in the table are, in each instance, those found when the mixture strength was that giving complete combustion.

The last column gives the calculated drop in temperature assuming no heat input and that the whole of the fuel be evaporated before entry to the cylinder.

Final Boiling Point.—It is always desirable to keep the final boiling point of any fuel as low as possible, because the higher boiling fractions are liable to condense on the cylinder walls and so to pass down into the crankcase, where they foul the lubricating oil.

As a general rule, so long as the final boiling point does not exceed say 400° F. no serious trouble need be feared on this score, for, if any fuel does condense on the walls, it will evaporate off in the crankcase, but in the case of kerosene and other high boiling fuels condensation on the cylinder walls and in the crankcase is one of the most serious difficulties with which designers of engines using these fuels have to contend.

Starting.—The readiness of a petrol to start from cold depends upon the proportion of low boiling-point fractions present in the fuel.

With nearly all commercial petrols the full vapour pressure at normal atmospheric temperature is only reached when at least 3 per cent of the volume of the vessel is occupied with liquid.

In an actual engine, starting is required with a minimum of about one-thirtieth of this, and even then the mixture strength would be about nine times richer than the normal running mixture. It follows, therefore, that unless a fuel has an abnormally high vapour tension it is necessary always to provide a very large excess for starting, e.g. by flooding or by the use of a special pilot jet, &c., as in the Zenith carburettor or by other means.

Calorific Value.—The heat liberated by the combination of the fuel and air is usually determined by burning the fuel in some form of calorimeter. The heat value so found includes the latent heat of the water formed, because in any form of calorimeter the temperature necessarily falls below the boiling point of water. Since, however, it is quite out of the question in any internal-combustion engine to make use of the latent heat of the water, it is customary to deduct from the total heat liberated that due to the condensation of the water formed. The heat value found after the deduction [illegible] ~~this deduction has been made,~~ is termed the lower calorific value of the fuel and is generally accepted as the basis upon which to calculate the thermal efficiency of an engine. In the case of internal-combustion engines using volatile liquid fuels, such a value for the available heat is not quite correct because, when a fuel is burnt in a bomb or other calorimeter, some of the heat of combustion is devoted to evaporating the remainder of the liquid fuel and is therefore absorbed. Now when used in an engine the whole of the liquid is evaporated before combustion takes place, and the heat required for its evaporation is supplied by the waste heat from the cycle or by the available heat already present in the air. In either case it is supplied by heat other than the useful heat of combustion of the fuel. If, therefore, it be accepted as correct that the latent heat of evaporation of the water formed should be deducted from the total heat of combustion as determined by calorimetric measurement, because this heat cannot be utilized, then it is equally right and proper that the latent heat of evaporation of the liquid fuel itself should be added to the observed calorimetric determination because its equivalent value in the heat of combustion can be and is used in any internal-combustion engine in which the fuel is evaporated before combustion starts, i.e. in any but Diesel engines; strictly speaking, there should be added the latent heat at constant volume, which is less than that at constant pressure by an amount equal to the work done on the atmosphere if the liquid is evaporated when unenclosed. In the following Table V is given the heat of combustion of various fuels in terms of B.Th.U.s per lb. and per gallon, the first two columns being the usually accepted lower calorific value and the second the revised value corrected to include the latent heat of evaporation of the liquid fuel itself. This latter corrected value is used throughout this volume for all determinations of thermal efficiency.

It cannot be too strongly emphasized that the heat value of a volatile liquid fuel bears no relation whatever to the power output

TABLE V

Fuel.	Calorific (lower) Value (exclusive of Latent Heat).		Calorific (lower) Value (including Latent Heat at Constant Volume).		Latent Heat of Evaporation of Fuel (at Constant Pressure atmospheric) B.Th.U. per lb.
	B.Th.U. per lb.	B.Th.U. per gallon.	B.Th.U. per lb.	B.Th.U. per gallon.	
Aromatic Free Petrol	19,080	137,000	19,200	136,200	133·0
“ A ” Petrol	18,450	144,300	18,580	145,200	142·0
“ B ” ,, ...	18,890	136,600	19,020	137,500	140·0
“ C ” ,, ...	19,000	138,100	19,120	137,000	135·0
“ D ” ,, ...	18,770	142,600	18,890	143,500	132·0
“ E ” ,, ...	18,970	136,400	19,090	137,100	132·0
“ F ” ,, ...	19,130	134,700	19,250	135,500	134·0
“ G ” ,, ...	...	...	...	...	...
“ H ” ,, ...	18,790	144,100	18,920	145,000	145·0
“ I ” ,, ...	...	...	...	...	...
Heavy Fuels					
Heavy Aromatics	17,900 (App.)	158,500 (App.)	18,030	159,600	136·0
Kerosene	19,000 (App.)	154,400 (App.)	19,100	155,200	108·0
Paraffin Series					
Pentane (Normal)	19,600	122,300	19,740	123,100	154·0
Hexane (80 % pure)	19,250	131,900	19,390	132,900	156·0
Heptane (97 % pure)	19,300	132,800	19,420	134,100	133·0
Aromatic Series					
Benzene (pure)	17,302	152,950	17,460	154,200	172·0
Toluene (99 % pure)	17,522	152,500	17,660	153,600	151·0
Xylene (91 % pure)	17,800	153,500	17,930	154,500	145·0
Naphthene Series					
Cyclohexane (93 % pure) ...	18,800	147,800	18,940	149,000	156·0
Hexahydrotoluene (80 %) ...	18,760	146,200	18,890	147,200	138·0
Hexahydroxylene (60 %) ...	18,770 (App.)	139,700 (App.)	18,890	140,600	133·0
Olefines					
Cracked Spirit (53 % unsat.) ...	18,400 (App.)	139,400 (App.)	18,540	140,200	150·0 (App.)
Alcohol Group, &c.					
Ethyl Alcohol (98 %)	11,480	91,600	11,840	94,500	406·0
,, ,, (95 vol. %) ...	10,790	88,000	11,130	92,000	442·0
Methyl Alcohol (Wood Naphtha)	9,630	79,900	10,030	83,300	500·0 (App.)
Methylated Spirits	10,200	83,700	10,580	86,900	450·0 (App.)
Butyl Alcohol (Coml.)	...	...	...	...	...
Ether (50 % in petrol)	16,700 (App.)	121,300 (App.)	16,830	122,500	146·0 (App.)
Carbon Disulph. (50 %) ...	10,600	105,400	10,730	106,600	146·0

obtainable from that fuel. In the case of gaseous fuels when the calorific value is very low, the apparent relationship is due largely to the presence in the gas of a considerable proportion of inert diluents such as nitrogen or carbon dioxide, in which case, of course, the power output is reduced because these diluents displace some of the available oxygen. In the case of volatile liquid fuels, however, the vapour contains no diluents at all, and the power output available is, therefore, entirely independent of the calorific value of the fuel.

The heat value of a fuel is, on the other hand, a direct measure of the quantity of fuel required, the lower the heat value the greater being the quantity needed to do the same work.

Heat Value of Mixture.—It is upon the heat value of the mixture of fuel and air, in the proportion required to give complete combustion, that the power output of an engine depends, and in this connection we find that all hydrocarbon fuels give within very close limits the same heat value per standard cubic inch of correct mixture. When allowance is made for the increase or decrease in specific volume after combustion, the variation becomes even less.

The following Table VI gives :—

Col. (1) The corrected calorific value of various petrols and other fuels.
Col. (2) The ratio of air/fuel by weight for complete combustion.
Col. (3) The increase or decrease in specific volume after combustion.
Col. (4) The ft.-lb. of energy liberated by the combustion of one standard cubic inch of mixture giving complete combustion, i.e. the total available internal energy.

The heat value of the " correct " mixture is usually termed the total " internal energy " of the working fluid, and this term will in future be used in order to distinguish it from the calorific value of the fuel, which latter has no influence on the power output.

Thermal Efficiency obtainable from Different Fuels.—Provided that the fuel is reasonably volatile, the thermal efficiency obtainable at any given compression ratio is substantially the same for all hydrocarbon fuels, irrespective of their chemical composition or of any other factor. In the case of the alcohol group, however, a somewhat higher thermal efficiency is obtained because, owing in part to their higher latent heat, and in part to their lower flame temperature, both the mean and the maximum temperatures of the cycle are lower, and the losses are therefore somewhat less. The range of burning on the weak side, which, by controlling the flame temperature, would control also the efficiency, happens to be almost

TABLE VI

Fuel.	(1) Calorific (lower) Value (including Latent Heat at Constant Volume).		(2) Air-to-Fuel Ratio by Weight for Complete Combustion.	(3) Spec. Volume after Combustion ÷ that before Combustion or "Volume Ratio."	(4) Total Energy liberated by Combustion per cub. in. at n.t.p. of Mixture giving complete Combustion foot-lb.
	B.Th.U. per lb.	B.Th.U. per gall.			
Aromatic Free Petrol	19,200	136,200	15·05	1·053	48·5
“ A ” Petrol	18,580	145,200	14·3	1·038	48·15
“ B ” ,,	19,020	137,500	14·7	1·049	48·45
“ C ” ,,	19,120	137,000	14·8	1·052	48·53
“ D ” ,,	18,890	143,500	14·6	1·047	48·35
“ E ” ,,	19,090	137,100	14·9	1·051	48·51
“ F ” ,,	19,250	135,500	15·0	1·053	48·54
“ G ” ,,	...	...	...	...	...
“ H ” ,,	18,920	145,000	14·7	1·048	48·31
“ I ” ,,	...	...	...	...	...
Heavy Fuels					
Heavy Aromatics	18,030	159,600	13·8	1·04	48·52
Kerosene	19,100	155,200	15·0	1·06	48·91
Paraffin Series					
Pentane (Normal)	19,740	123,100	15·25	1·051	48·7
Hexane (80 % pure)	19,390	132,900	15·2	1·051	48·35
Heptane (97 % pure)	19,420	134,100	15·1	1·056	48·64
Aromatic Series					
Benzene (pure)	17,460	154,200	13·2	1·013	47·51
Toluene (99 % pure)	17,660	153,600	13·4	1·023	47·98
Xylene (91 % pure)	17,950	154,500	13·6	1·03	48·1
Naphthene Series					
Cyclohexane (93 % pure)	18,940	149,000	14·7	1·044	48·11
Hexahydrotoluene (80 %)	18,890	147,200	14·7	1·047	48·2
Hexahydroxylene (60 %)	18,890	146,600	14·8	1·054	48·59
Olefines					
Cracked Spirit (53 % unsat.)	18,540	140,200	14·8 (App.)	1·054	49·54
Alcohol Group, &c.					
Ethyl Alcohol (98·5 %)	11,840	94,500	8·9	1·065	47·39
,, ,, (95 vol. %)	11,130	92,000	8·4	1·065	46·86
Methyl Alcohol (Wood Naphtha)	10,030	83,300	6·5	1·06 (App.)	48·2 (App.)
Methylated Spirit	10,580	86,900	8·0 (App.)	1·064	48·82
Butyl Alcohol (Coml.)	...	...	...	...	...
Ether (50 % in petrol)	16,830	122,500	13·0	1·06	49·2
Carbon Disulph. (50 %)	10,730	106,600	10·8	0·98	39·4

exactly the same in the case of all volatile liquid fuels yet examined with the exception of ether, and, in all cases, the maximum thermal efficiency is obtained when the mixture is 15 per cent weak. If any attempt be made to weaken the mixture beyond this point, combustion becomes unduly slow and incomplete, and the efficiency falls away in consequence. Theoretically, of course, the efficiency should rise steadily as the mixture strength is reduced and should follow an almost straight line law until at the point when the mixture is infinitely weak the efficiency should be equal to the air standard. In practice the extent to which the mixture can be weakened with increase of efficiency depends to some small extent, in the case of a single-cylinder engine, upon the position of the sparking plug and the intensity of the spark, while in a multi-cylinder engine it depends to a much larger extent upon the uniformity of distribution, but in both cases it is very limited.

It is perhaps rather remarkable that the maximum thermal efficiency obtainable with two fuels so widely different as hexane C_6H_{14} and benzene C_6H_6 should be the same, but the explanation lies in the fact that while for CO_2 the dissociation is great at high temperatures, yet the increase in specific heat is small. On the other hand, for H_2O the dissociation is small but the increase in specific heat is great. Curiously enough, these effects almost exactly balance one another, so that the sum of the losses from each source is practically the same.

The following Table VII gives :—

Col. (1) The lower calorific value of the fuel corrected for the latent heat of evaporation.
Col. (2) The lowest fuel consumption in terms of lbs. and pints per I.H.P. hour at a compression ratio of 5 : 1.
Col. (3) The corresponding thermal efficiency.

The above figures relate to the thermal efficiency obtainable when all fuels are used at the same compression ratio.

It has, however, been stated previously that the highest compression ratio at which a fuel may be used depends upon its tendency to detonate, and it has been shown that this varies widely.

The following Table VIII shows the fuel consumption in terms both of pints and lbs. per I.H.P. hour when each fuel is used at its highest useful compression ratio.

It will be noted that in the case of kerosene and one or two other examples which are not very volatile, the observed thermal efficiency is considerably lower, due to the fact that a substantial

TABLE VII

Fuel.	(1) Calorific (lower) Value (including Latent Heat at Constant Volume).		(2) Minimum Consumption at Compression Ratio of 5 : 1 per I.H.P. Hour.		(3) Thermal Efficiency at Compression Ratio of 5 : 1 per cent.
	B.Th.U. per lb.	B.Th.U. per gallon.	lb.	Pints.	
Aromatic Free Petrol	19,200	136,200	0·415*	0·462*	31·9*
" A " Petrol	18,580	145,200	0·432	0·442	31·7
" B " ,, ...	19,020	137,500	0·423	0·468	31·7
" C " ,, ...	19,120	137,000	0·421	0·463	31·6
" D " ,, ...	18,890	143,500	0·422	0·445	31·9
" E " ,, ...	19,090	137,100	0·421*	0·469*	31·7*
" F " ,, ...	19,250	135,500	0·414	0·471	31·9
" G " ,, ...	...	...	0·426	0·454	...
" H " ,, ...	18,920	145,000	0·425	0·443	31·7
" I " ,, ...	...	...	0·418	0·460	...
Heavy Fuels					
Heavy Aromatics	18,030	159,600	0·510	0·461	27·6
Kerosene	19,100	155,200	0·523*	0·515*	25·4*
Paraffin Series					
Pentane (Normal)	19,740	123,100	...	...	...
Hexane (80 % pure)	19,390	132,900	0.411	0·480	32·0
Heptane (97 % pure)	19,420	134,100	0·410*	0·475*	31·9*
Aromatic Series					
Benzene (pure)	17,460	154,200	0·458	0·415	31·8
Toluene (99 % pure)	17,660	153,600	0·455	0·418	31·7
Xylene (91 % pure)	17,930	154,500	0·452	0·420	31·4
Naphthene Series					
Cyclohexane (93 % pure) ...	18,940	149,000	0·420	0·427	31·9
Hexahydrotoluene (80 %) ...	18,890	147,200	0·425	0·430	31·7
Hexahydroxylene (60 %) ...	18,890	140,600	0·424*	0·456*	31·8*
Olefines					
Cracked Spirit (53 % unsat.) ...	18,540	140,200	0·429	0·453	32·0
Alcohol Group, &c.					
Ethyl Alcohol (98 %)	11,840	94,500	0·663	0·665	32·4
,, ,, (95 vol. %) ...	11,130	92,000	0·705	0·692	32·5
Methyl Alcohol (Wood Naphtha)	10,030	83,300	0·777	0·750	32·7
Methylated Spirits	10,580	86,900	0·740	0·721	32·5
Butyl Alcohol (Coml.)	...	...	0·566	0·550	
Ether (50 % in petrol)... ...	16,830	122,500	...	...	...
Carbon Disulph. (50 %) ...	10,730	106,600	...	...	...

* This sign indicates that the values are only calculated, since these fuels could not be tested at a compression ratio of 5 : 1 owing to detonation. The values have been inserted to show the efficiency and power obtained relatively to the other fuels if used at the same compression.

TABLE VIII

Fuel.	Minimum Consumption at Highest Useful Compression per I.H P. Hour.	
	lb.	Pints.
Aromatic Free Petrol	0·422	0·471
"A" Petrol	0·393	0·402
"B" ,,	0·393	0·435
"C" ,,	0·410	0·451
"D" ,,	0·407	0·428
"E" ,,	0·435	0·484
"F" ,,	0·412	0·469
"G" ,,	0·449	0·478
"H" ,,	0·389	0·405
"I" ,,	0·457	0·503
Heavy Fuels		
Heavy Aromatics	0·447	0·404
Kerosene	0·581	0·571
Paraffin Series		
Pentane (Normal)	..	..
Hexane (80 % pure)	0·405	0·473
Heptane (97 % pure)	0·491	0·568
Aromatic Series		
Benzene (pure)	0·392	0·355
Toluene (99 % pure)	0·385	0·354
Xylene (91 % pure)	0·381	0·354
Naphthene Series		
Cyclohexane (93% pure)	0·385	0·392
Hexahydrotoluene (80 %)	0·394	0·404
Hexahydroxylene (60 %)	0·429	0·461
Olefines		
Cracked Spirit (53 % unsat.)	0·405	0·428
Alcohol Group, &c.		
Ethyl Alcohol (98 %)	0·532	0·533
,, ,, (95 vol. %)	0·565	0·555
Methyl Alcohol (Wood Naphtha)	0·725	0·700
Methylated Spirits	0·625	0·609
Butyl Alcohol (Coml.)	0·472	0·459
Ether (50 % in petrol)	..	..
Carbon Disulph. (50 %)	..	..

proportion of the liquid fuel is deposited on the walls of the induction piping and cylinder and escapes combustion. This proportion could be reduced by further pre-heating, but all the tests were carried out, for comparative purposes, at exactly the same heat input to the carburettor.

By additional pre-heating a slightly higher thermal efficiency can be obtained, but the power output is reduced and the tendency to detonate increased thereby.

The Maximum Power Output.—The maximum power output obtainable from any fuel depends upon the internal energy of the working fluid and upon the latent heat of evaporation of the liquid.

The former varies very little, the latter considerably as between different fuels—generally speaking as regards the true hydrocarbon fuels the variations in internal energy and latent heat just about balance, with the result that the maximum power output is the same for all. For example, the total internal energy of benzene is about 1·5 per cent less than that of hexane, on the other hand the latent heat of benzene is considerably greater, and a greater weight of mixture is therefore retained in the cylinder, with the result that under identical temperature and other conditions both give, at the same compression ratio, the same power output to within less than half of 1 per cent.

In the case of alcohol, although the total internal energy of the mixture is appreciably lower, yet the latent heat is so much greater that a much denser charge is retained in the cylinder and the power output is some 5 per cent greater despite the lower internal energy.

Table IX shows :

Col. (1) The total internal energy.
Col. (2) The latent heat of evaporation.
Col. (3) The observed indicated mean pressure (power output) at a compression ratio of 5 : 1.
Col. (4) The observed indicated mean pressure at the highest useful compression ratio.

Fuels for Aircraft.—For all commercial purposes other than aircraft, fuel is supplied by bulk, not weight, and it is therefore the heat value per gallon and not per lb. which need be considered. In the particular case of aircraft, however, the weight and not the bulk of the fuel becomes the primary consideration. Other things being equal, therefore, the fuel with the highest calorific value per lb. will be the most efficient. From this point of view alone, the paraffin series would appear the most desirable. Unfortunately, however, owing to their tendency to detonate, the members of this series cannot be used in a high compression and therefore in an efficient engine.

TABLE IX

Fuel.	Total Energy liberated by Combustion per cub. in. of Mixture giving complete Combustion foot-lb.	Latent Heat of Evaporation of Fuel (at Constant Pressure atmospheric) B.Th.U. per lb.	Max. Ind. Mean Effective Pressure at Compression Ratio of 5 : 1. Heat $\frac{65 \text{ B.Th.U.}}{\text{Min.}}$ lb. per sq. in.	Max. Ind. Mean Effective Pressure at Highest Useful Compression. Heat $\frac{65 \text{ B.Th.U.}}{\text{Min.}}$ lb. per sq. in.
Aromatic Free Petrol	48·5	133·0	131·3*	130·0
“ A ” Petrol	48·15	142·0	131·2	140·1
“ B ” ,,	48·45	140·0	131·5	137·5
“ C ” ,,	48·53	135·0	131·0	133·9
“ D ” ,,	48·35	132·0	131·2	134·9
“ E ” ,,	48·51	132·0	131·0*	128·6
“ F ” ,,	48·54	134·0	131·8	132·7
“ G ” ,,	. .	. .	131·5*	127·4
“ H ” ,,	48·31	145·0	131·0	139·5
“ I ” ,,	. .	. .	131·7*	125·1
Heavy Fuels				
Heavy Aromatics	48·52	136·0	131·1	142·5
Kerosene	48·91	108·0	130·7	123·0
Paraffin Series				
Pentane (Normal)	48·7	154·0	131·3	139·0
Hexane (80 % pure)	48·35	156·0	132·3	133·1
Heptane (97 % pure)	48·64	133·0	131·2*	119·5
Aromatic Series				
Benzene (pure)	47·51	172·0	131·6	146·5
Toluene (99 % pure)	47·98	151·0	131·5	147·0
Xylene (91 % pure)	48·1	145·0	131·5	146·8
Naphthene Series				
Cyclohexane (93 % pure)	48·11	156·0	131·3	139·0
Hexahydrotoluene (80 %)	48·2	138·0	131·0	137·9
Hexahydroxylene (60 %)	48·59	133·0	130·8*	130·0
Olefines				
Cracked Spirit (53 % unsat.)	49·54	150·0 (App.)	131·6	136·0
Alcohol Group, &c.				
Ethyl Alcohol (98 %)	47·39	406·0	137·8	156·5
,, ,, (95 vol. %)	46·86	442·0	142·0	161·2
Methyl Alcohol (Wood Naphtha)	48·2 (App.)	500·0 (App.)	144·8	146·6
Methylated Spirits	46·82	450·0 (App.)	144·5	155·5
Butyl Alcohol (Coml.)	. .	. .	138·0	156·0
Ether (50 % in petrol)	49·2	146·0 (App.)	136·0*	125·0
Carbon Disulph. (50 %)	39·4	146·0	124·5	125·7

*** This sign indicates that the values are only calculated, since these fuels could not be tested at a compression ratio of 5 : 1 owing to detonation. The values have been inserted to show the efficiency and power obtained relatively to the other fuels if used at the same compression.**

If the compression were adjusted to suit the fuel, then the greatest distance could be flown on that fuel which gives the highest value for the product thermal efficiency × heat value per lb., and it is interesting to note that of all the fuels examined the highest figures are:

	Efficiency per cent.		Heat Value. B.Th.U. per lb.		Product.
(1) Xylene	0·373	×	17,930	=	6700
(2) Cyclohexane	0·349	×	18,940	=	6600
(3) Petrol (sample B) ..	0·341	×	19,020	=	6500

Although xylene comes first on the list, it is not a practicable fuel on account of its high boiling point and low volatility. Cyclohexane, one of the lighter members of the naphthene series, would actually give the best possible results from every point of view, but is not obtainable in bulk. The next on the list, viz. the petrol sample (B), is a very light and highly volatile fuel which was prepared by the Asiatic Petroleum Co. for the cross-Atlantic flight of 1919.

Since that time the fuel technologist has been at work and there has been notable progress in the production of fuels with increasing resistance to detonation, but still retaining the essential characteristics of low boiling point and high heat value. In consequence the compression ratio of aircraft engines, then slightly under 5 : 1, has risen to 7 : 1 to-day, while even at these high ratios a considerable supercharge can be tolerated with present-day aviation petrol.

In aircraft engines it is particularly desirable to use a very volatile fuel, both on account of distribution and in order to be able to reduce to a minimum the amount of pre-heating required, and so to make the maximum possible use of the latent heat of evaporation to increase the power output.

On these counts, alcohol, of high latent heat value, would seem to be an excellent fuel. As is seen from Table X, however, it has a relatively low heat value and a high specific gravity—drawbacks to its general use in circumstances where specific fuel consumption is all-important, as controlling the weight of fuel to be carried.

For special purposes, such as racing cars and aircraft in which the power output for short sprints is the deciding factor, full use of its high latent heat of evaporation can be made.

The method adopted in 1919 of expressing the detonation tendency of a fuel in terms of "toluene value" has been mentioned. It was found difficult, however, to obtain toluene of sufficient purity to serve as an absolute standard, since a very small proportion of certain impurities difficult to eradicate greatly affected its anti-detonating properties. Since the question of fuel rating had assumed such world-wide importance it was essential to use for this purpose chemically pure substances which could be matched exactly and made available in all countries.

In 1931 the Co-operative Fuel Research Committee of the American Society of Engineers, acting in co-operation with the Institution of Petroleum Technologists, reviewed the whole question. The upshot of their work has been that toluene has given place to iso-octane, while *n*-heptane has replaced the original aromatic-free petrol. A special C.F.R. research engine has been adopted as the standard apparatus for measuring the " octane number " of fuels, and the method of determination has been standardized.

To-day the tendency of a fuel to detonate is expressed in terms of its " octane number ", which, as a measure of fuel quality, is now generally accepted as having greater significance than any other characteristic. By the scientific blending of fuels from various sources—natural and synthetic—and by the addition in varying quantity of lead tetra-ethyl, benzol, alcohol, &c., the octane number of fuels has risen year by year, allowing a corresponding increase in engine power output.

Here a warning is necessary. The word " octane " used in the expression " octane number " refers to iso-octane, a compound of high anti-knock quality, *not* to the member of the paraffin series bearing the same name (see Table on p. 5).

Some further notes on octane rating will be given in Chapter II.

The corresponding classification number for Diesel fuels is the " cetane number ", the implications of which are dealt with in the chapter on the High-speed Diesel Engine.

Summary

The results of a lengthy investigation of the various volatile liquid fuels carried out on behalf of the Asiatic Petroleum Co. may be summarized as follows:—

(1) It has been proved that the tendency of a fuel to detonate is the one outstanding factor in determining its value for use in a constant-volume internal-combustion engine. Compared with this, most other considerations are of secondary importance.

(2) There appears to be little doubt as to the correctness of the view, now generally accepted, that detonation is largely dependent

upon the normal rate of burning of the fuel and is less the lower the rate of burning.

(3) In all cases it seems that a low rate of burning is advantageous. No fuel has yet been found whose rate of burning was too low to permit of maximum efficiency being obtained in the highest speed engine yet tested.

(4) Fuels capable of standing a very high compression will operate in a low-compression engine, equally as efficiently as those whose normal rate of burning is high—provided that there is a reasonable degree of turbulence in the combustion chamber.

(5) Apart from the limitations introduced by detonation, the power output obtainable from all volatile liquid fuels, with the exception of the alcohol group, is the same at the same compression to within less than 2 per cent. Such variations as occur, within this range, are due rather to variations in the latent heat of evaporation than to any other circumstance.

(6) Owing to the high latent heat and low boiling point of alcohol and certain other bodies, the weight of charge per cycle is greater and a higher power output is obtained in consequence.

(7) The efficiency with which all volatile fuels, other than alcohol, &c., are burnt is practically the same, at the same compression ratio, irrespective of rate of burning, provided the compression is low enough to avoid detonation under any circumstances. In the case of alcohol, the efficiency is slightly higher, on account of the lower flame temperature.

(8) The useful range of burning is, to all intents and purposes, the same for all volatile liquid fuels.

(9) The unavoidable losses due to the combined influence of dissociation and change of specific heat at high temperatures are substantially the same in all cases.

(10) All the experimental results indicate that the performance of any combination of hydrocarbons as regards detonation, and therefore the power output and efficiency obtainable, is the mean performance of each of the components. The performance of any complex fuel such as petrol can therefore be predicted, once the nature and proportion of its constituents are determined, or conversely, a fuel can be prepared to give any required performance, within the limits available.

(11) The highest useful compression ratio for, and therefore the power output obtainable from, any petrol is governed by the relative proportions of aromatics, naphthenes, and paraffins it contains—the

smaller the proportion of the latter the better from almost every point of view.

(12) To judge of the quality of a fuel by its specific gravity is entirely misleading. If naphthene and aromatic fractions are present in any large proportion (as is frequently the case), then a high specific gravity is a substantial advantage.

(13) Owing to the very rich mixture delivered normally by pilot jets, and the still further enrichment effected by flooding, only a relatively small proportion of highly volatile constituents appears to be required for starting.

Table X (facing p. 30) and Table XI (facing p. 32) give a general summary of the above investigations and test results which were carried out on behalf of the Asiatic Petroleum Co. That the author is permitted to publish them without reservation of any kind is due to the generosity and public spirit of this most enterprising company. The following particulars of the single-cylinder research engine and its accessories which formed the principal piece of apparatus used for the research, may be of some interest:—

In figs. 3–6 are shown drawings and photographs of the variable compression engine referred to in connection with the preceding tests.

In the design of this engine the following considerations were taken into account:—

(1) In view of the prolonged and extensive nature of the tests, not only were durability and reliability regarded as matters of primary importance, but every effort was made to ensure mechanical consistency.

(2) Every expedient known at the time, viz. 1919, was adopted to attain the highest possible thermal efficiency and power output, and to ensure that all losses, whether thermal or frictional, were reduced to the absolute minimum, and maintained as nearly constant as possible, under all conditions.

(3) The engine was designed to run when required at a piston speed in excess of that of contemporary engines.

(4) Means are provided for varying the compression of the engine over any range from 3·7 : 1 up to 8 : 1 while running at full power, and without disturbing any temperature, frictional, mechanical, or other conditions.

(5) The combustion chamber is so designed that its general form and ratio of surface to volume undergo the minimum of alteration when the compression is varied, and to this end a very long stroke is employed.

(6) Special means are adopted to render the engine as little sensitive to changes in the temperature of the lubricant as possible. Ball bearings are used wherever possible, in order to reduce variation in friction with different oil temperatures, and the water jacketing round the barrel of the cylinder is stagnant, and therefore quickly attains a constant temperature, independent of the temperature of the supply. This ensures that the piston friction, which is dependent upon the temperature of the oil on the cylinder walls, reaches a minimum in the course of a few minutes, and thereafter remains constant. The importance of retaining as nearly as possible the same general form of combustion chamber under all conditions of compression cannot be overestimated. Very misleading results have been obtained when the compression ratio has been varied by fitting different pistons, some with concave, others with convex crowns. In one series of experiments with different compression ratios which the writer examined, and which were obviously carried out with the most scrupulous care, the results were entirely vitiated because the whole character and efficiency of the combustion chamber were completely changed, as between the low compression and the high, with the result that a certain optimum compression ratio was claimed to have been found, after which any further increase in compression resulted in loss of power and efficiency. A careful scrutiny of the results showed that at or near the so-called optimum compression ratio the efficiency of the combustion chamber was at a maximum, and that at the higher compressions it fell away very rapidly, and, indeed, became quite exceptionally inefficient.

In the variable compression engine designed for the purpose of these tests, the efficiency of the combustion chamber undergoes very little change between the lowest compression ratio and the highest, with the result that the efficiency increases with increase of compression at a perfectly steady rate throughout the whole range.

As will be seen from the sectional drawings, figs. 3 and 4, the compression ratio is varied by raising or lowering the whole cylinder, together with the carburettor, camshaft, and valve gear (fig. 5); by this means the compression ratio can be varied over any range in the course of a few seconds, and without disturbing any of the temperature conditions or any adjustments.

To measure and record the compression ratio in use, a micrometer is provided as shown in fig. 6. This is arranged to operate electrical contacts, and controls a pilot lamp, which lights up immediately the desired compression is reached, so that the operator

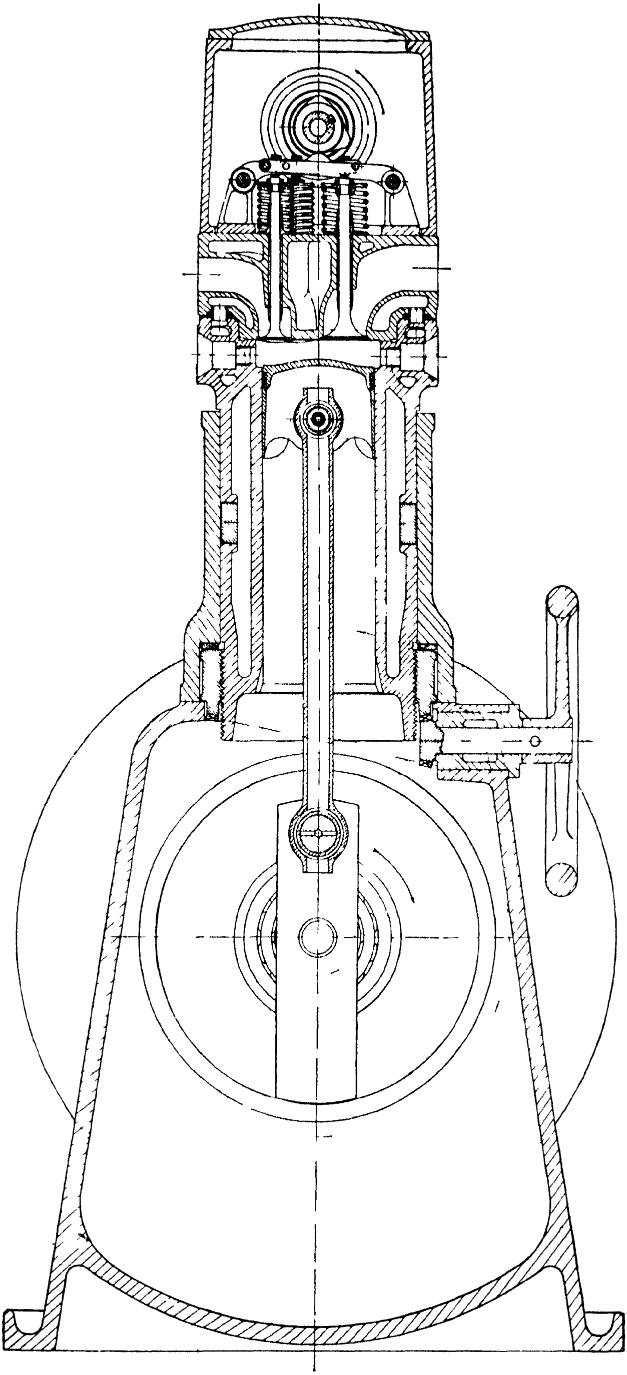

Fig. 3

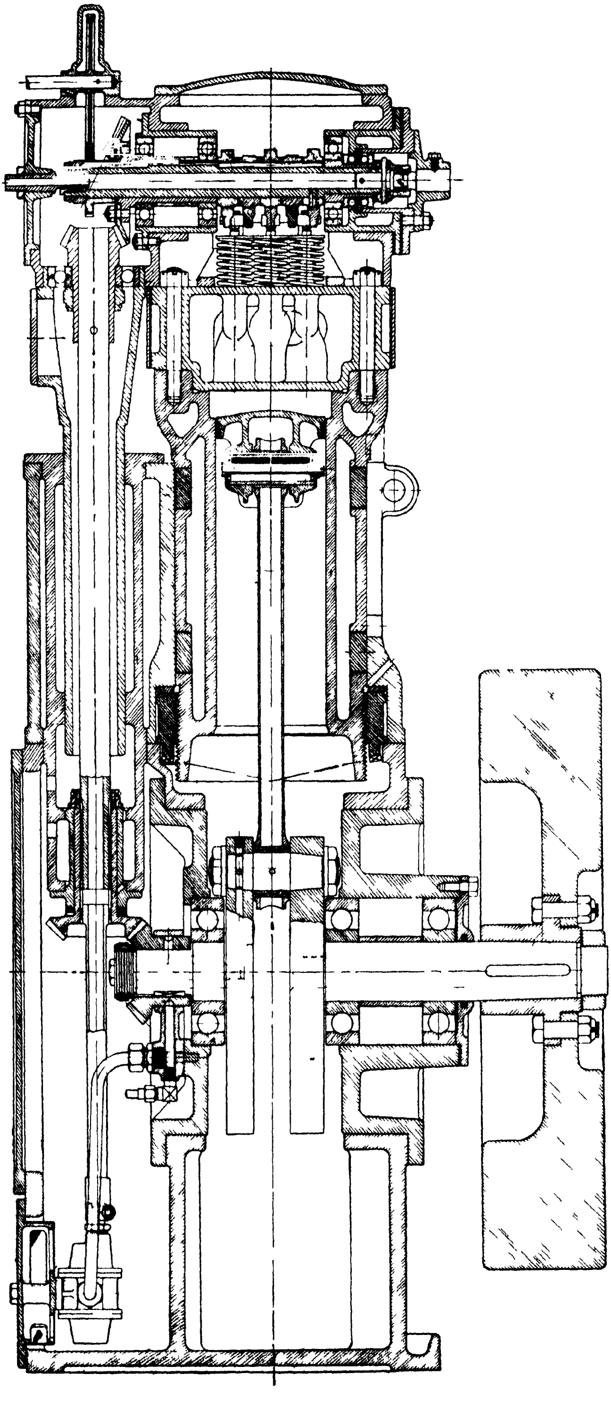

Fig 4.—Variable Compression Engine, Sectional Side Elevation

can adjust the micrometer screw at his leisure to the compression ratio he requires before making any alteration, and can then see at a glance by the lighting up of the lamp that this compression has been reached.

For the ignition of the charge four sparking plugs are fitted equidistant round the circumference of the combustion chamber, each of which is connected to a Remy high-tension coil. The low-tension circuit of all the coils is operated by a single Remy contact breaker driven directly from one end of the camshaft. The object of using this arrangement in preference to magnetos was twofold:

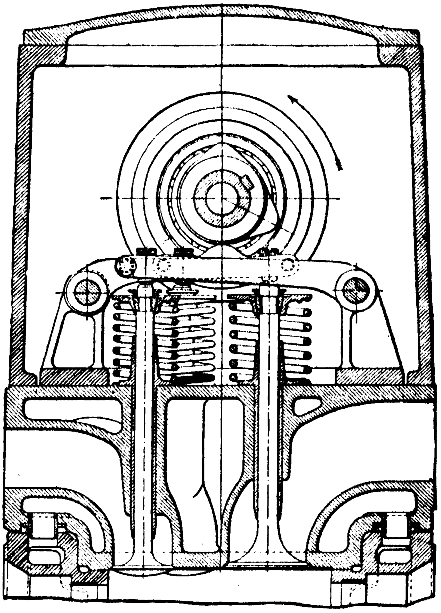

Fig. 5.—Section through Cylinder Head showing Valve Gear

(1) To ensure that the passage of the spark across all four plugs should be absolutely synchronous.

(2) To ensure that the intensity of the spark should be the same at all settings.

In practice, it was found that the use of two sparking plugs on opposite sides of the combustion space gave equally good results, and all tests were therefore run under these conditions. To measure accurately both the power and friction losses the engine is direct coupled to a balanced swinging field electric dynamometer, one arm of which carries a dead weight of 40 lb., which is slightly in excess of the maximum torque of the engine—a light open-scale spring balance is used to record the difference in torque between the dead weight and that developed by the engine—this arrangement permits of exceedingly accurate determinations, since a very small variation in torque corresponds with a wide range on the spring balance. The mean torque on the dynamometer arm is in the neighbourhood of 35 lb., and the difference can be read off at a glance to within less than one-tenth of a pound. The steadiness of the dynamometer is such that the needle of the spring balance does not vibrate or oscillate through a range of more than ±0·1 lb. Generally speaking, all readings of torque may be taken as being accurate to within one-third of 1 per cent, while the standard of accuracy of the average of several readings is, of course, considerably higher. The load is controlled by varying the field excitation of the dynamometer. For this

Fig. 6.—Photo of Upper Part of Engine showing Micrometer for measuring Compression Ratio

purpose, two rheostats are provided in the field circuit, one of which gives coarse graduations, and the other, a continuous coil resistance, affords continuous range, and is used for fine adjustments.

The fuel-measuring device consists of two vessels, each comprising two conical-ended chambers connected together at either end by a narrow throat, as shown in fig. 7. The upper chamber has a capacity of exactly one pint, and the lower of ¼ pint. A gauge glass is fitted to each vessel. The rate of fall of liquid in the glass is very rapid when passing the narrow throats, so that its passage past the marks on the gauge glass can be clocked with extreme accuracy. Geared off one end of the camshaft is a revolution counter operated

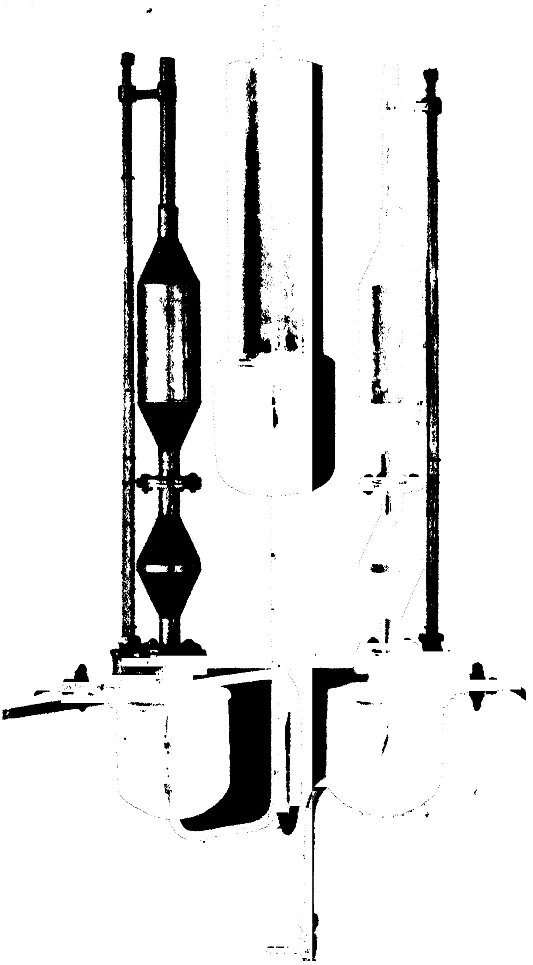

Fig. 7.—Fuel-measuring Device

by means of a magnetic clutch, and so arranged that the counter is thrown into operation as the liquid in the gauge glass passes the first mark, and is thrown out, and a brake applied to its spindle as the liquid passes the second mark; thus the actual number of revolutions during the consumption of either one pint or $\frac{1}{4}$ pint of fuel is automatically recorded.

The carburettor is a standard Claudel-Hobson aircraft type, but fitted with a fine adjustment needle valve controlling the jet, so that the mixture can be varied between close limits. An electrical heater is fitted in the carburettor air intake passage, and the exact amount of heat supplied can be read off from instruments on the switchboard. A thermometer fitted in an insulated pocket, and projecting into the inlet valve port, is provided to record, very approximately, the temperature of the working fluid during its entry to the cylinder. From the known amount of heat supplied, and from the measured difference in the temperature of the air before and after its entry to the carburettor, it is possible to determine at least a relative measure of the mean volatility of the fuel used. The readings of the thermometer in the induction passage are, however, of relative value only. Owing to the variations in the temperature of the thermometer pocket due to the deposition of liquid fuel upon it, this thermometer behaves as a wet bulb instrument, and even the relative values of its readings cannot be relied upon as between fuels of widely different latent heats of evaporation.

A general lay-out of the apparatus is shown in fig. 8.

A small calibrated gas-holder is also provided, from which the engine can draw its supply of air when running on a liquid fuel.

The fall of the gas-holder controls an electrical contact mechanism which in turn operates a magnetic counter on the observer's desk; this counter being inter-connected, electrically, with the revolution counter on the engine. By this means the air consumption per revolution can be measured with great accuracy.

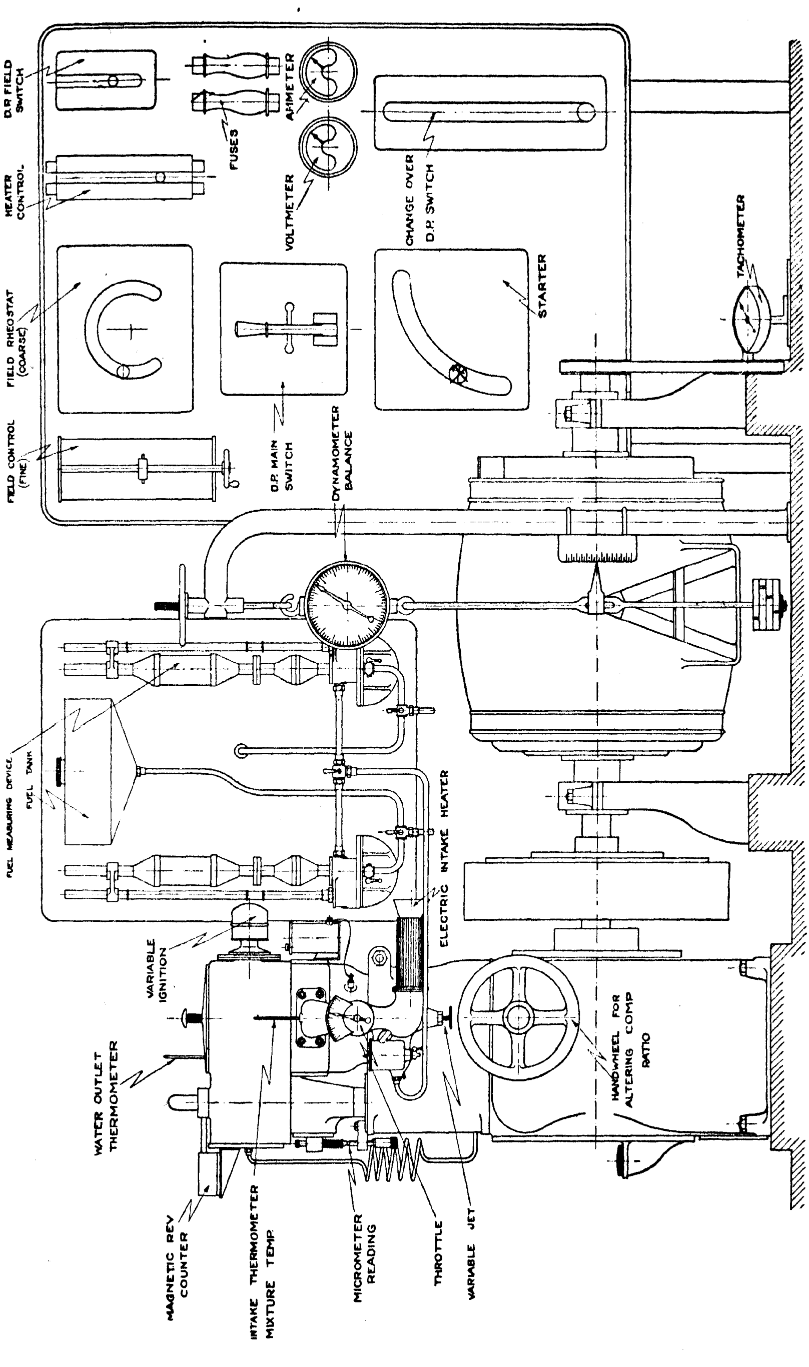

Fig. 8.—General Lay-out of Testing Apparatus

CHAPTER II

DETONATION

The phenomenon of detonation appears to be the setting up in the cylinder of an explosion wave. This occurs when the rapidity of combustion of that portion of the working fluid first ignited is such that, by its expansion, it compresses before it the unburnt portion beyond a certain rate. When the rate of temperature rise due to compression by the burning portion of the charge exceeds that at which it can get rid of its heat by conduction, convection, &c., by a certain margin, the remaining portion ignites spontaneously and nearly simultaneously throughout its whole bulk, thus setting up an explosion wave which strikes the walls of the cylinder with a hammer-like blow and, reacting in its turn, compresses afresh the portion first ignited. This further raises the temperature of that portion, and with it the temperature of any isolated or partially insulated objects in its vicinity, thus ultimately giving rise to pre-ignition. It would appear, therefore, pretty certain that detonation depends primarily upon the rate of burning of that portion of the charge first ignited, and it remains to discover what actually controls this rate.

This view of the mechanism of detonation, formulated many years ago and based on tests carried out in the "E.35" variable-compression engine and other units, has been confirmed in its essential implications by more recent research.

The discovery was made by Midgley in 1922 that lead tetraethyl had a power of suppressing detonation in an engine in a most remarkable manner. This led other investigators to examine the problem, notably Callendar and Egerton and his associates, whose research work has reached such a point that a closer peep behind the scenes is now possible.

Very briefly it may be said that the condition of things giving rise to detonation depends on a process of chemical change, which begins relatively slowly during the compression stroke and continues with increasing rapidity as the temperature rises. Certain unstable compounds are formed during this time in a series of chain reactions, and those compounds which take part in the violent combustion responsible for the " knock " are produced. It appears that, concurrently with this process, organic peroxides are formed, probably in small quantity. These peroxides are a vital factor in the reaction, and their formation appears to be dependent on rise of temperature, either by compression and radiation from the advancing flame front, or on the catalytic action of a hot surface. There is a suspicion that nitrogen peroxide, too, plays a part in the process.

Detonation, then, is due to the violent reaction between certain unstable partial products of combustion formed in the unburnt portion of the charge, small quantities of peroxide compounds and, of course, the oxygen present in the charge. While the main combustion is in progress conditions become favourable and a greater or smaller proportion of the remainder of the charge ignites and burns with great rapidity, generating very high local pressure and temperature. Such detonation may occur in more than one part of the combustion chamber, and the quantity of gas thus burnt often varies widely from cycle to cycle.

Influence of the Nature of the Fuel upon Detonation.— This subject has already been touched on in Chapter I. Broadly speaking, it would appear, both from actual engine tests and from Tizard's researches, that two factors determine whether or not a fuel will detonate:

(1) The self-ignition temperature of the fuel/air mixture.

(2) The rate of acceleration of burning as the ignition temperature is exceeded.

Both the true self-ignition temperature, if indeed such a term can be used, and the rate of acceleration of burning appear to depend primarily upon the chemical composition of the fuel.

Reference has been made to the self-ignition temperature of a fuel/air mixture. Strictly speaking there is no such thing, for any fuel/air mixture will actually combine in time at almost any temperature. In the case of hydrocarbon fuels the rate of combination is multiplied about three times for every 3 per cent increase in temperature, so that, in practice, there is a comparatively narrow range of temperature over which the rate of burning is reasonable. In this connection the case of carbon disulphide is particularly interesting— this fuel has an exceedingly low ignition temperature, so low that when used neat in an engine it will pre-ignite at once even at a compression ratio of 3·6 : 1, yet, when mixed with petrol, carbon disulphide is quite effective in preventing detonation and will allow of the use of a higher compression ratio than is normally possible with petrol. On investigation it has been found that, in the particular case of carbon disulphide, the rate of acceleration of burning is very much slower and that it trebles its rate for every 7 per cent rise in temperature as against about every 3 per cent for all hydrocarbon fuels.

Apart from the variations in the composition of the fuel, various design factors enter also into the question and exert a powerful influence. Of these the most important appears to be the absolute distance from the point of ignition to the most remote corner of the combustion chamber. This, however, is not the whole story, for the tendency to detonation is influenced also by the degree of turbulence of the gases in the remote corner and by the temperature of the boundary surfaces, both of which factors determine the facility with which the entrapped gases can dispose of the heat generated by local compression in front of the advancing inflammation. Thus the tendency to detonation will be greatest when the corner most remote from the sparking plug contains relatively stagnant gas or is bounded by some hot surface, such as the head of the exhaust valve.

Since the absolute length of flame path appears to be the main determining factor, it follows that, given similarity of design, the smaller the cylinder, the higher the ratio of compression which can be employed with any given fuel. From this consideration it appears that the efficiency of the petrol engine will increase as the size of the cylinder is reduced. Against this, however, both the heat loss to the cylinder walls and the loss due to the proportion of chilled and incompletely burned gas increases as the cylinder size is reduced.

Effect on Engine Efficiency and Power.—The net result of these various factors is that the thermal efficiency of a petrol engine is, within wide limits, nearly independent of its size. The specific power output of a small engine is greater than that of a large one, since the cycle can be performed more frequently. The reason for this difference is purely a mechanical one, the limits being due to the inertia stresses in the moving parts.

In the smaller engine the conditions are more favourable to higher speeds of rotation. In these circumstances, however, the frictional losses are generally greater than in a larger engine of the same output running at a lower rotational speed. This difference in friction is relatively quite small, so that while the brake horsepower (per unit of its volume) of the small cylinder is much greater than that of the large one, the brake thermal efficiency of the small engine is only slightly, if any, less.

Examples of Effect of Cylinder Size and Design.—As illustrating the very wide difference in the compression ratio which can be used in different engines on the same fuel, the following comparative tests with two variable compression engines, both highly efficient, may be quoted.

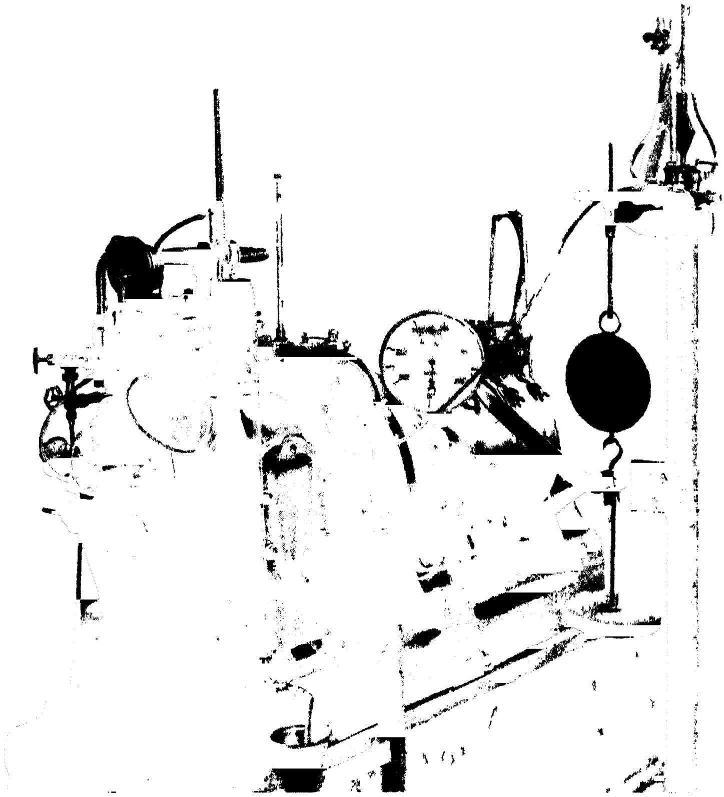

Fig 9

One engine, " E.35 ", has a cylinder bore of 4·5 in. and a stroke of 8 in. with five overhead poppet valves (two inlet and three exhaust) described in Chapter I. Ignition is by two synchronized sparks occurring at diametrically opposite sides of the combustion chamber, the distance between the two sparking plugs being 5 in. exactly. With a certain fuel and with the mixture strength and ignition timing adjusted to give maximum power, the highest useful compression ratio (i.e. the highest useful compression as limited by *the incidence of detonation) is found to be 5·3.* The ratio corresponds with an air standard efficiency of 48·5 per cent.

The second engine, "E.5", figs. 9, 10, and 11, has a cylinder bore of 2·75 in. and a stroke of 3·25 in., with a single sleeve valve.

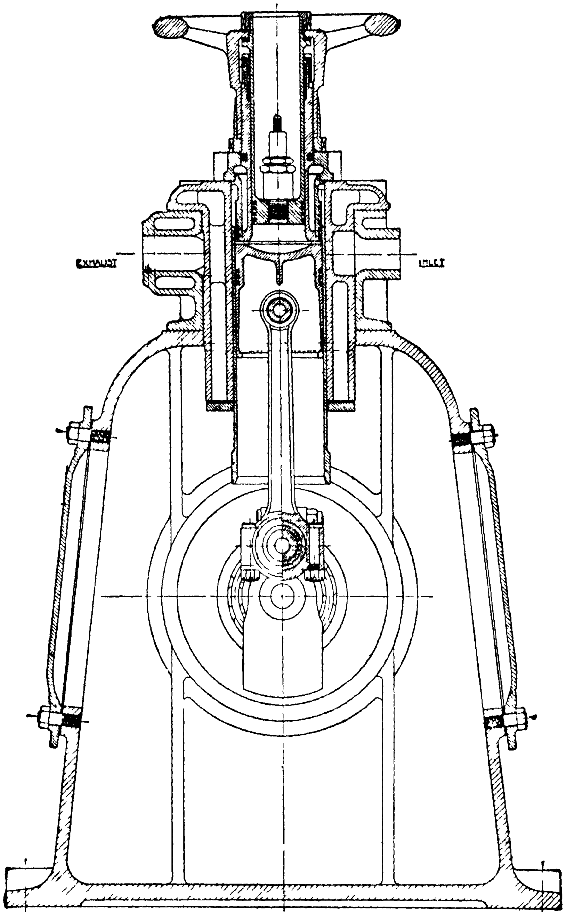

Fig. 10.—Cross Sectional Arrangement of Single-sleeve-valve and Research-unit Type—E.5-1

Ignition is by a single plug placed centrally in the combustion chamber, which is of slightly conical form, the extreme distance from the ignition point to the furthest point in the combustion chamber being 1·7 in. With the same fuel and under the same conditions as to mixture strength and ignition advance, the highest useful compression, as limited by detonation, is found to be 6·8. This ratio corresponds to an air standard efficiency of 53·5 per cent.

Thus, the difference between these two engines due to the combined influence of design factors and scale effects is no less than 1·5 ratios of compression, corresponding with an increase in thermal efficiency

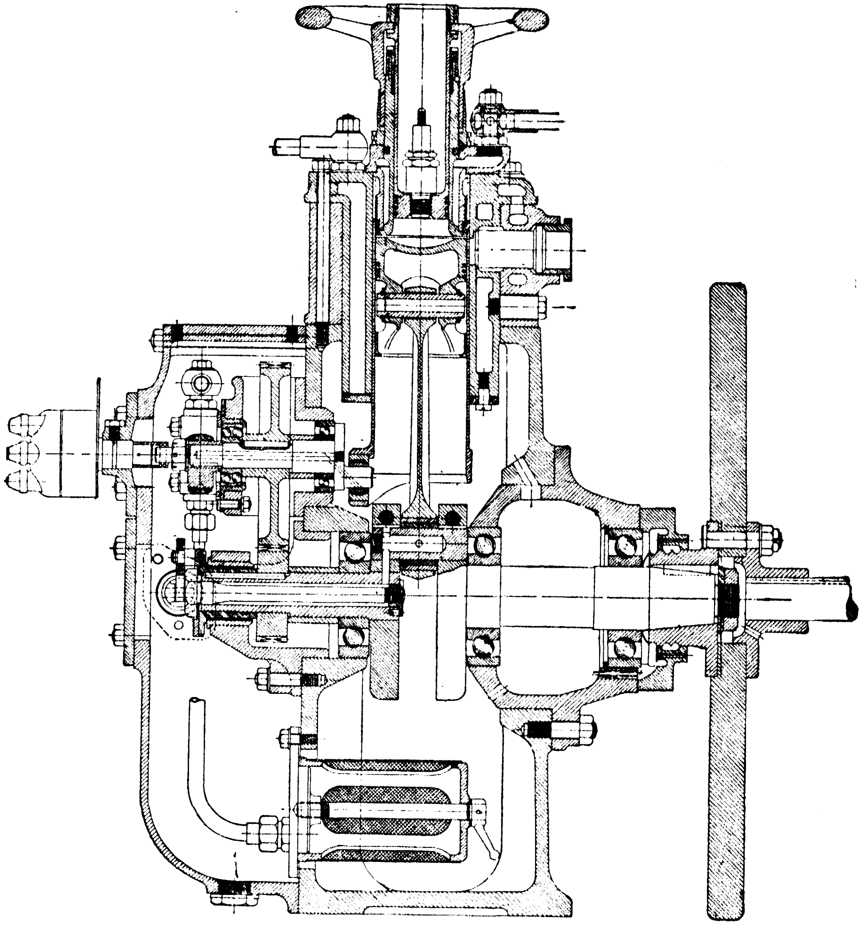

Fig. 11.—Single-sleeve-valve Research-unit Type—E.1-5

of 10 per cent. Here the difference in favour of " E.5 " is attributable primarily to:—

(*a*) The smaller cylinder and therefore reduced length of flame travel.

(*b*) The absence of hot exhaust valve heads.

Taking another extreme variation in design, a " T-headed " poppet-valve engine of 4·375-in. bore and 6-in. stroke, in which the sparking-plug was placed directly over the inlet valve and therefore at one side, detonated heavily on the same fuel at a compression ratio of only 3·7.

As showing the effect of cylinder size, apart from differences of general design, may be quoted the results of tests made with four experimental single-cylinder single-sleeve-valve engines. The bores of these engines are 2·75, 3·5, 5·5, and 8·5 in. respectively. All have very nearly the same stroke/bore ratio and the same form of combustion chamber, namely, a cone with the sparking-plug at the apex. They are, in every respect, as nearly similar as is possible over so large a range of sizes. In each case also the maximum length of flame path was approximately 25 per cent greater than the cylinder radius.

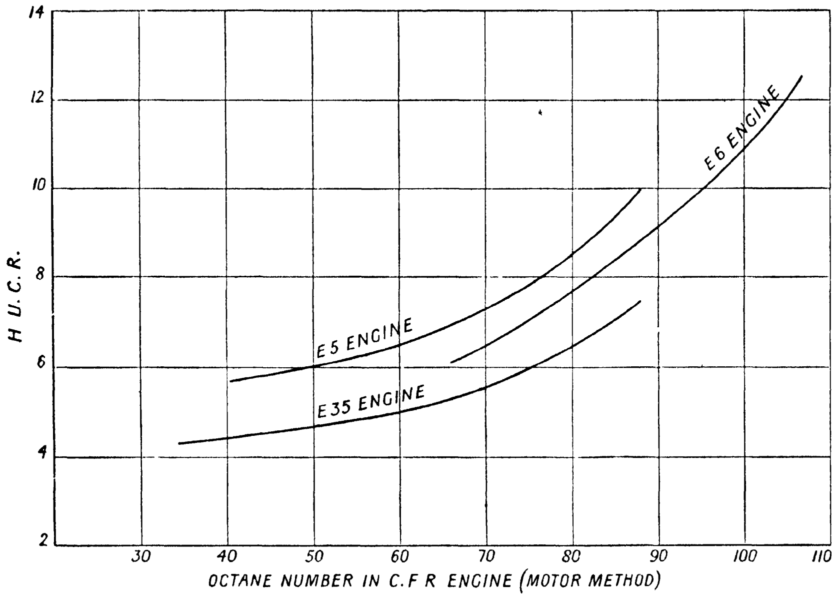

Fig. 12.—Curves showing Relationship between H.U.C.R. and Octane Number

The highest useful compression ratio as determined by detonation for each of these engines was very carefully observed and may be expressed for the same fuel (20 per cent/80 per cent benzol-petrol blend) as follows:—

Diameter of cylinder (in.)	2·75	3·5	5·5	8·5.
H.U.C. Ratio	7·9	7·5	6·2	5·4.

In each case the engine speed was adjusted to that giving maximum torque, and the temperature conditions, mixture strength, and all other relevant factors were made as nearly as possible the same for each engine.

The observations, when plotted against cylinder size, will be found to lie on a reasonably smooth curve, except that the 5·5-in. cylinder is relatively rather low in H.U.C.R. This is probably due to some difference in the degree or nature of the turbulence of the charge prior to ignition.

Example of the Effect of Fuel.—Despite these wide differences between engines due, in part, to design factors and, in part, to scale effect, the relative tendency to detonation as between different fuels is nearly the same for different engines. Fig. 12 shows the H.U.C.R., or highest useful compression ratio, for the two engines previously described, " E.35 " and " E.5," with different fuels for which the octane number in a C.F.R. engine (Motor Method) has been determined. To the curves taken on these two engines has been added a third, that from a small variable-compression poppet-valve research unit, " E.6," bore 3 in., stroke $4\frac{3}{8}$ in., designed to enable fuels of higher octane rating to be tested by the H.U.C.R. method. This engine, which can be run as either a petrol or a Diesel unit by interchanging the cylinder head and by alternatively fitting a magneto or a fuel pump to a universal platform, is shown in figs. 13 and 14 fitted with a Diesel cylinder head, and in fig. 15 as a research unit for volatile fuels.

The points of interest in the curves shown in fig. 12 are three. In the first place, it is clear that three engines of very different design and size, while they give very different absolute H.U.C.R. values with the same fuel, have similar relative tendencies to detonate with fuels of different quality.

Secondly, the curves illustrate the profound influence of engine design on detonation. Considering the range of fuels in the figure common to all three engines, octane numbers 65 to 87, we see that the difference in H.U.C.R. between the " E.5 " sleeve-valve unit and the " E.6" poppet-valve unit is no less than 0·8 ratio in favour of the sleeve-valve, while the difference in cylinder diameter between the two engines is only 0·25 in. On the other hand, the corresponding differences between the two poppet-valve engines " E.6 " and " E.35 " are 1·0 ratio and 1·5 in. respectively.

Thirdly, the curves give a general idea of the compression ratios permissible in practice with the improved-quality fuels available to-day, and the increase in compression ratio which may be used by stepping up the octane rating of the fuel. Since 1921 the octane

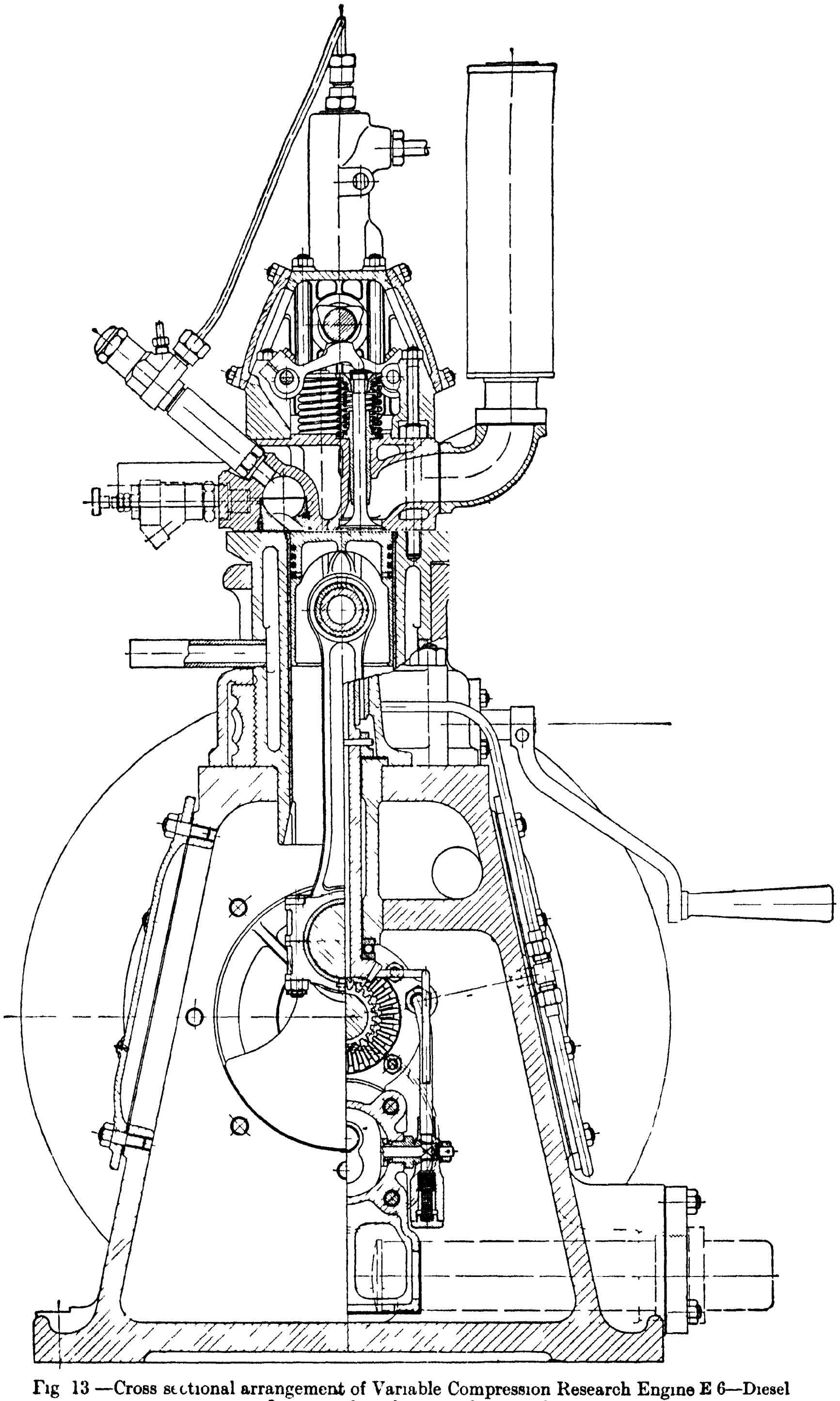

Fig 13 —Cross sectional arrangement of Variable Compression Research Engine E 6—Diesel Version 3 in bore × 4$\frac{1}{2}$ in stroke

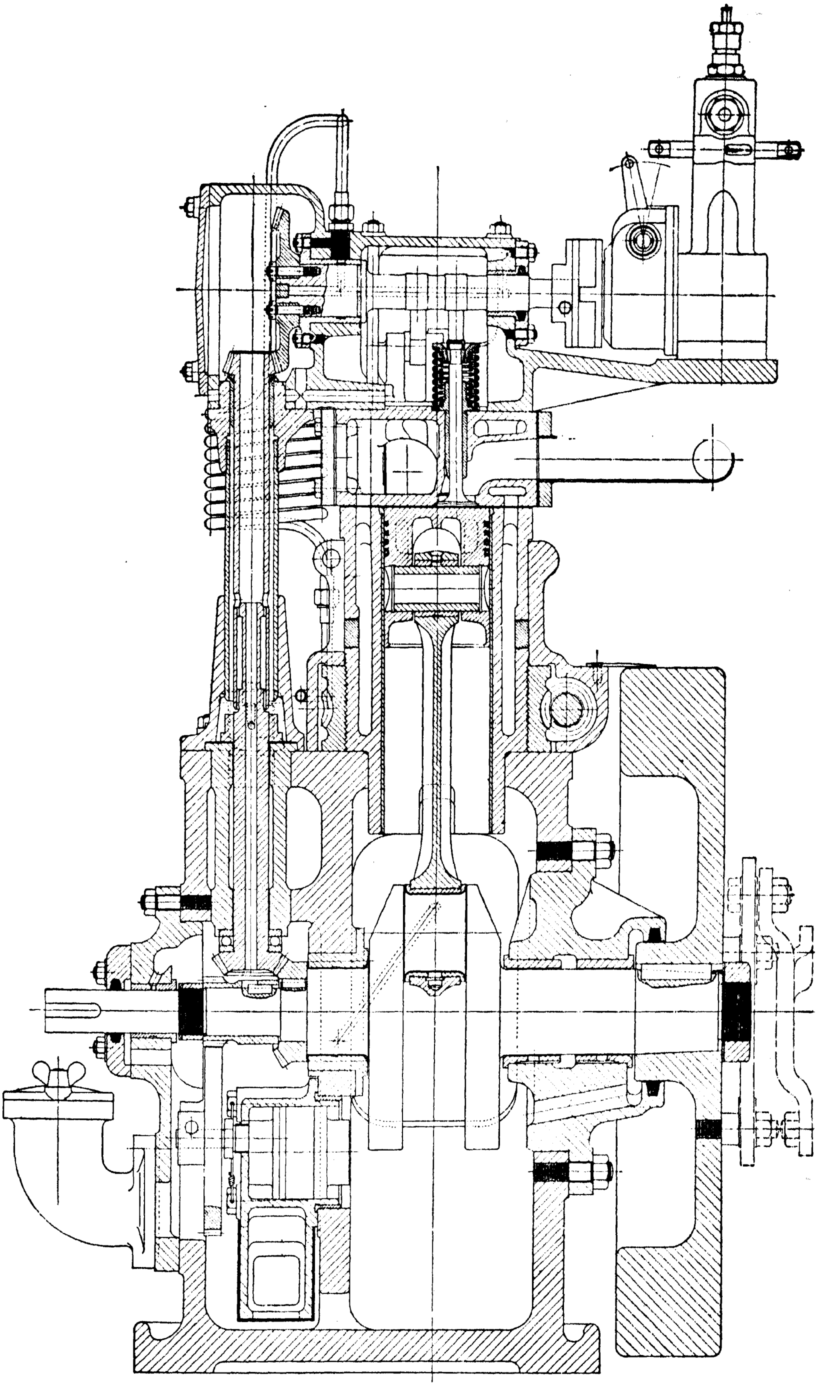

Fig. 14.—Longitudinal arrangement of Variable-Compression Research Engine E.6—Diesel Version. 3 in. bore × 4⅜ in. stroke

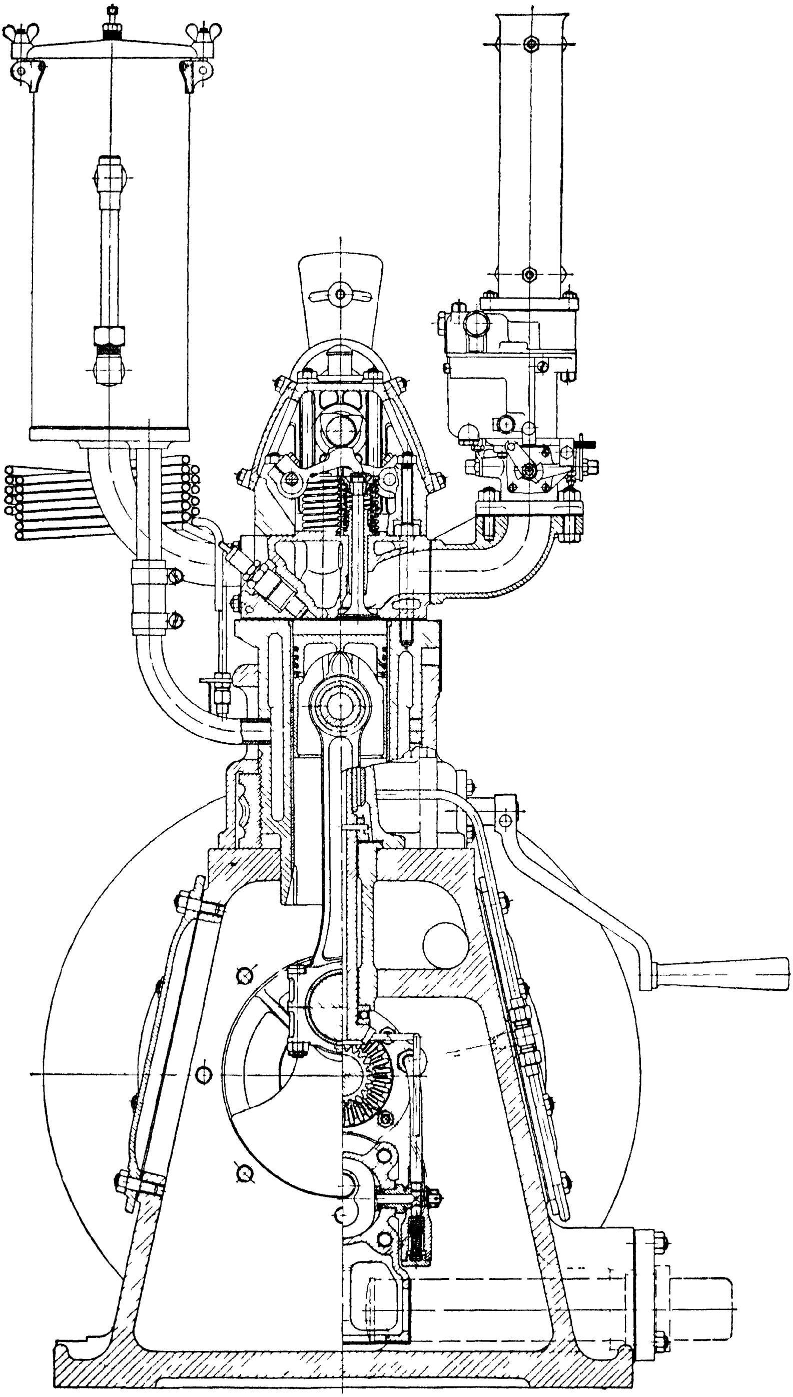

Fig. 15.—Cross-sectional arrangement of Variable-Compression Research Engine E 6—Petrol Version 3 in. bore × $4\frac{3}{8}$ in. stroke

number of No. 1 motor spirit has risen from approx. 50 to 70 or 72 to-day. From the curves it is seen that this improvement enables the designer to use a compression ratio from 1 to 2 ratios higher, according to the size and design of the cylinder. In point of fact, over this period the average compression ratio for automobile engines has risen about 1¾ ratios.

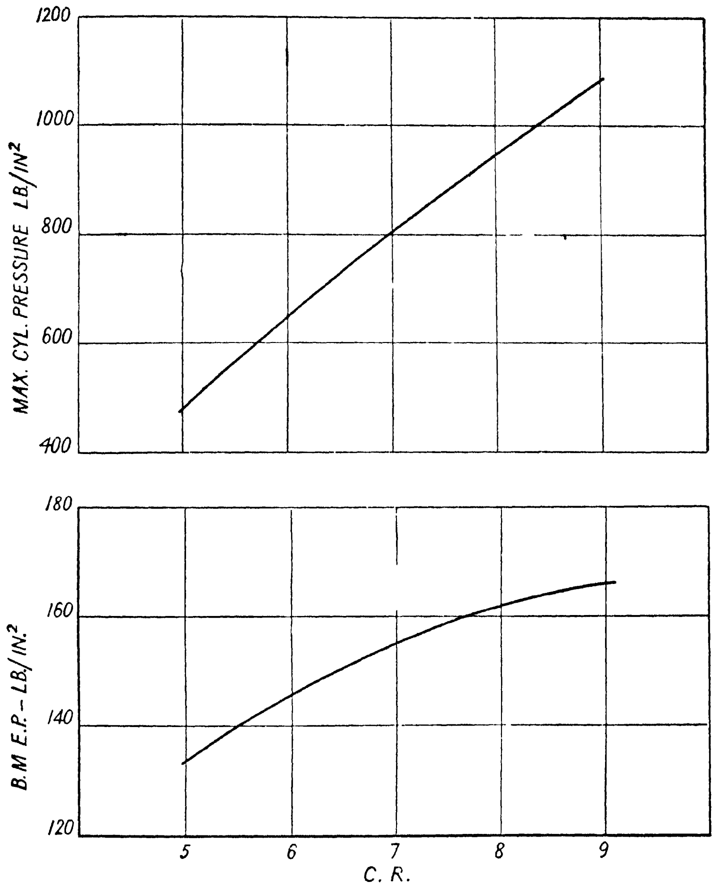

Fig. 16.—Curves of Brake Mean Pressure and Maximum Cylinder Pressure, plotted against Compression Ratio

The curves emphasize the fact that although the engine design factor exercises a great influence on the tendency to detonation, it has little or no effect on the relative H.U.C.R. of different fuels.

At first sight the prospect of increasing engine performance by utilizing the improved anti-knock qualities of fuels to increase the compression ratio is a very attractive one, but as this process is pursued it becomes less and less attractive. As we proceed to the higher compression ratios the maximum pressure increases at an alarming rate, while the corresponding gain in power output becomes disappointingly small. This point is illustrated by the curves shown in fig. 16, where the rise in maximum cylinder pressure and in mean pressure is plotted against a range of compression ratios embracing those in general use to-day.

Example of the Effect of Position of the Point of Ignition.—As showing the influence of sparking-plug position on detonation, the following very carefully conducted series of tests may be cited as generally representative. The engine used for the purpose

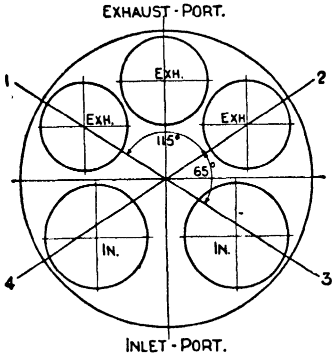

Fig. 17.—Sketch showing Valve and Sparking-plug Positions. Positions 1, 2, 3, and 4.

of the tests was the "E.35" variable compression engine. This engine is provided with four radial plug positions and special means exist to ensure that any two plugs can be fired simultaneously with perfect synchronism. The position of the plugs and their relation to that of the valves is indicated in diagram fig. 17. Throughout the full range of observations the following conditions were maintained constant:—

Speed 1500 R.P.M.
Mixture strength 15 per cent rich (maximum power setting).
Jacket temperature 50° C.
Heat input to the induction system 1350 watts.
Fuel, "Texas gasoline".

Upon analysis these results show clearly that the ignition advance required and the H.U.C.R. are both dependent very greatly upon the length of flame path from the sparking-plug to the farthest point reached by the flame. The relative positions of the

TABLE I

Plug Position.	Ignition Timing giving Max. Torque.	H.U.C.R.	Brake Mean Effective Press. lb. per sq. in.
1 and 3	30° early	5·3	138·0
2 and 4	30° "	5·25	136·6
3 and 4	34° "	4·98	134·5
1 and 2	32° "	4·9	133·8
1 and 4	32° "	4·8	132·1
2 and 3	32° "	4·84	132·8
1 only	39° "	4·95	133·5
2 "	39° "	4·9	133·1
3 "	40° "	4·84	132·8
4 "	42° "	4 85	132·8

active sparking-plugs and of the various valves are shown to exert a secondary influence as well.

Examples of the Effects of Mixture Strength and Timing of Ignition.—The influence of mixture strength on

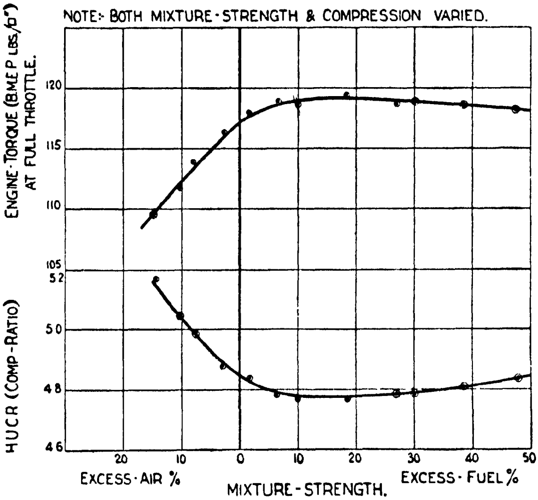

Fig. 18

H.U.C.R. and engine output in the case of the single-cylinder engine in which the issue is not confused by errors in uniformity of distribution is shown graphically in fig. 18. In fig. 19 is shown the influence of mixture strength on the ignition advance required to give maximum output. In this test, the compression ratio was fixed at a point below that at which detonation could intrude.

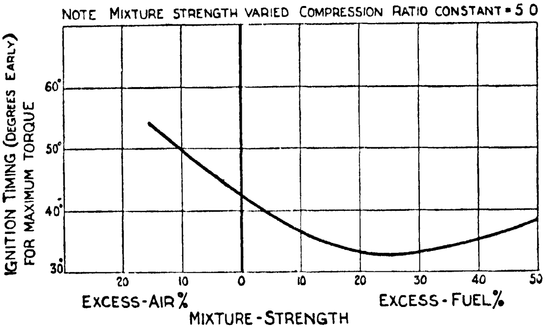

Fig. 19.—" E.35 " Engine

• Fig. 20 shows the ignition advance required and the engine output obtained as the compression ratio was varied, all the conditions being as nearly constant as possible. Curves from the two poppet-

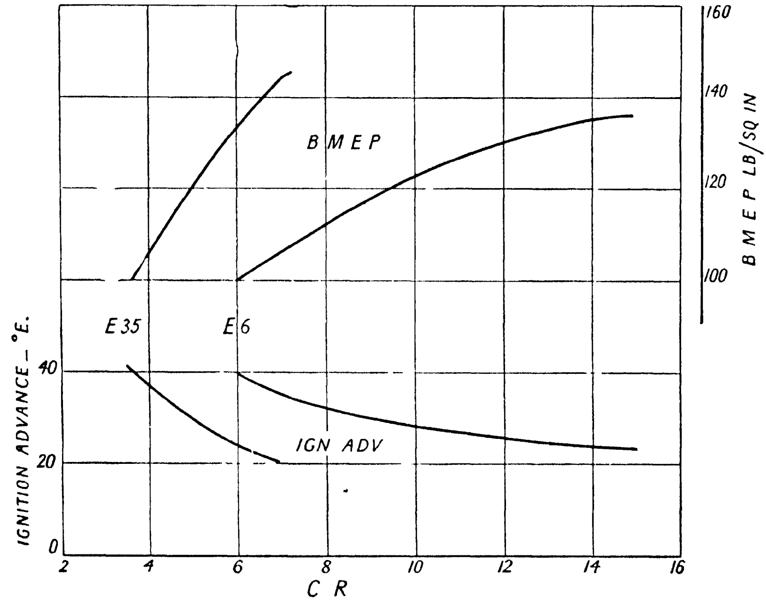

Fig. 20

valve units "E.35" and "E.6" are given. The flattening of the engine torque curve as the compression ratio reaches the higher values should be noticed. Another point of interest is the depressing effect on the brake torque of the higher frictional losses of the smaller unit, which was designed with very large bearing surfaces and with very heavy scantlings to deal with the high maximum pressures developed at high compression ratios—up to 20 : 1.

Detonation and the Engine.—Some consideration has been given to the effect of various features in engine design in promoting detonation in an engine. This aspect of the problem will receive further treatment when the influence of combustion-chamber design is considered in Chapter IV.

Here a few words may be devoted to the reverse influence, the effect of detonation on the running and behaviour of an engine.

Detonation signifies its occurrence by the " knock " heard by the human ear when pressure waves of very high local pressure break against the walls of the combustion chamber. If allowed to persist it may be the cause of grievous harm to the engine and lead to serious depreciation in the power output and efficiency.

In engines required to operate for considerable periods at high output, such as a modern aero-engine, detonation which is barely audible may lead to the piston-rings sticking in their grooves, to high-temperature corrosion of the pistons or, in more aggravated cases, to actual cracking and break-up of the pistons. In engines which do not work at such a high duty, e.g. the motor-car engine, detonation must be very severe to interfere seriously with the otherwise normal running of the engine or its useful life.

Broadly speaking, as the power output has risen, the intensity of heat flow to the inner surfaces has increased. The result is that the speed of combustion and the working temperatures of the internal surfaces have been raised to a higher level, and if the phenomenon of detonation is superimposed on combustion in these conditions, the danger of serious internal damage to the engine is all the greater.

So long as detonation is not so severe as to involve such a large proportion of the charge that jacket heat loss becomes excessive, the actual effect on the power developed is relatively small. This is understandable when it is realized that in normal circumstances detonation affects the burning of only a small proportion of the charge. Fig. 21 shows the loss in performance over the speed range due to detonation which just disappeared on the inferior fuel at 1800 R.P.M. With the better fuel there was no detonation at any speed. It is seen that at speeds below 1200 R.P.M., detonation is very severe on the poorer fuel and the power loss more serious.

A feature of the running of the engine which is detonating heavily is that uncombined carbon emerges in the exhaust as puffs of black smoke which by observation can be seen to correspond in degree with the intensity of the knock. It seems that owing to the extreme rapidity of the reaction the local temperature of the centre of the gas thus detonated is such that complete combination of carbon with oxygen (such as occurs in normal combustion) becomes impossible.

Another effect follows if the centre of detonation happens to be in a relatively confined space close to the edge of the piston. The nucleus of intensely high pressure and temperature pit-marks the pistons by a high-temperature corrosion process which may, in prolonged conditions of detonation, literally eat away the piston material. Cast-iron resists this corrosion to a far greater extent than aluminium. In the case of an aluminium piston which had run for 50 hours under detonating conditions, at one spot in the region of detonation the piston had been eaten into to a depth of $\frac{1}{8}$-in., while all round this spot was evidence of local corrosion in the form of deep pitting. It appears that the centre of detonation, while restricted to one part of the chamber, is not confined within very close limits, but may occur over an area say 1 or 2 in. wide, depending on the irregularities of the flame-spread, and perhaps a variation in the quantity and location of the partial products taking part in the reaction.

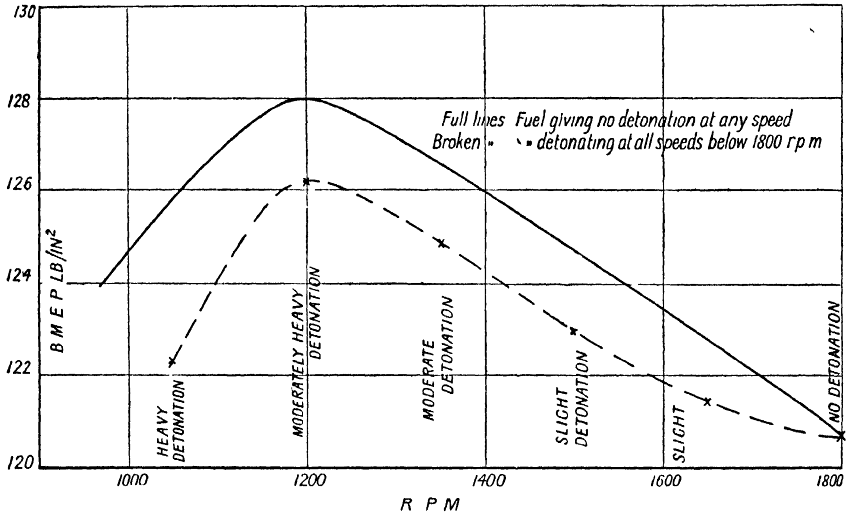

Fig. 21

Again, the effects of detonation may show themselves internally not so much in corrosion of the piston as by causing decomposition of the lubricating oil in the ring grooves into a gummy substance which, assisted by the free carbon which is present, befouls and sticks the rings in their grooves.

In general, it seems that detonation gives rise to very high temperature and pressure of a strictly local character and the surfaces attacked are either those which run hot because of their unfavourable location in regard to cooling, or those which, by the restricted character of their environment, are close to the centre of detonation, that is to say, close to those gases which are at the highest temperature and pressure.

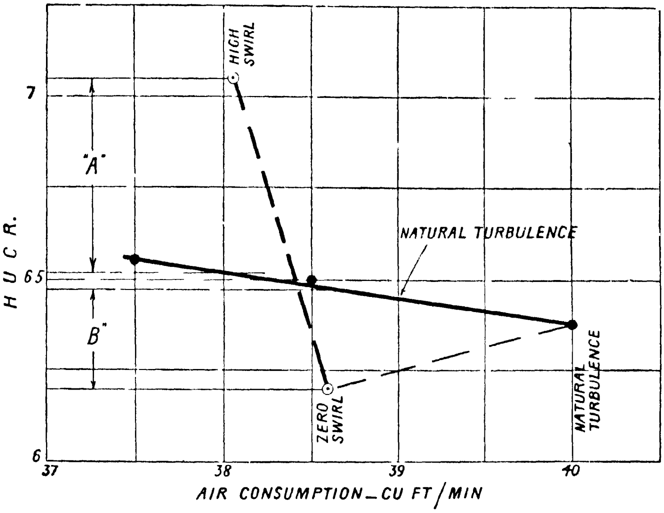

Fig. 22.—Effect of Air Movement on H.U.C.R.

Effect of air movement on H.U.C.R.—It was mentioned on p. 42 that the tendency to detonate is influenced by design factors such as the turbulence of the gases in the remote corner. Fig. 22 shows the effect on H.U.C.R. of varying the air turbulence by fitting a shrouded inlet valve to the variable-compression engine (*a*) to give a high degree of swirl, (*b*) to give a condition of zero swirl. The operation of shrouding the valve adversely affects the volumetric efficiency, and in order to show the effect of increased air swirl, two curves of H.U.C.R. are plotted against air consumption. The full line shows the H.U.C.R. from full to slightly reduced throttle opening with natural turbulence, a standard inlet valve being used without shroud. The broken line includes two readings of H.U.C.R., one with a high degree of swirl and one at zero air swirl. It is seen that with increased swirl a gain of over 0·5 ratio in H.U.C.R. is realized at the same volumetric efficiency ("A "). On the other hand, with swirl reduced to a value less than the normal turbulence through the valves, there is a reduction in H.U.C.R. of 0·3 ratio at the same volumetric efficiency (" B "). In this case the valve was shrouded, in a way found from blowing tests on models, to kill the natural movement of air generated by passage through the inlet valve.

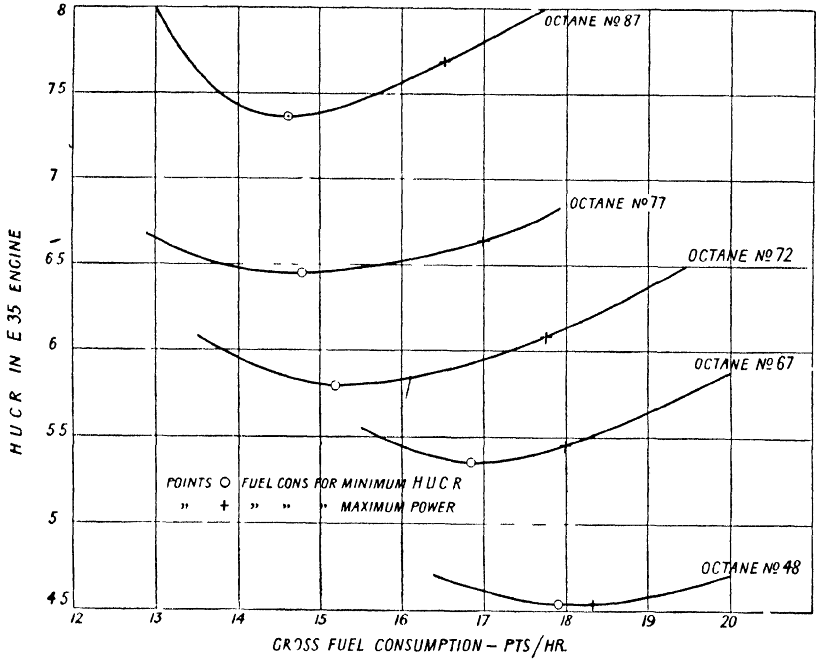

Fig. 23.—Effect of Mixture Strength on H.U.C.R.
Points ⊙ Fuel Consumption for Minimum H.U.C.R.
Points + Fuel Consumption for Maximum Power

H.U.C.R. in relation to Mixture Strength.—Fig. 18 showed how the H.U.C.R. varies with mixture strength on a given fuel. Fig. 23 shows that while the general character of the curve of H.U.C.R. against mixture strength remains the same for a wide range of anti-knock value, the higher the octane number, broadly speaking, the weaker the mixture strength at which maximum detonation occurs.

The results plotted in these groups of tests were obtained with " E.35 ", the overhead-valve type variable compression engine.

Metallic Anti-Knocks.—The remarkable power of suppressing detonation possessed by lead tetra-ethyl and other metallic compounds is best illustrated by stating that one molecule of lead tetra-ethyl in 50,000 molecules of mixture (fuel and air) may raise the H.U.C.R. by one ratio.

It is not proposed to discuss fully the many fantastic theories that have been proposed to account for the action of metallic "anti-knocks". Callendar's admirably reasoned explanation and the theory proposed by Egerton must, however, be referred to.

Briefly, Callendar's theory suggests that the active agents in promoting detonation are the unstable peroxides which are formed as intermediate products during the process of combustion of hydrocarbons. The presence of certain readily oxidizable metals in a finely-divided state provides an alternative and more attractive partner for the oxygen and thus prevents the formation of peroxides as products of partial combustion. According to Callendar's theory, the progress of events is briefly as follows:—The metal compound, which is volatile and soluble in petrol, is carried into the cylinder as a vapour along with the fuel and air. Under the conditions of temperature and pressure existing within the cylinder during the early stages of compression, the unstable metallic compound dissociates, setting free the metal in a very finely divided and highly active state. It is assumed, then, that at least a portion of the fuel during compression is still in a liquid but finely-divided state, and that each droplet forms a nucleus for oxidation. It is at this stage, during the formation of organic oxides or peroxides, that the metal probably intervenes, to reduce these compounds. It is presumed that, owing to the instability of such organic peroxides in very small proportions, their presence increases greatly the speed of subsequent combustion and, therefore, the consequent detonation. The theory would explain the fact that so little metal is needed to suppress detonation. Moreover, this theory will serve to explain other anomalies as between undoped fuels of different physical characteristics.

Still more recently Egerton has proposed that the action of "anti-knocks" is somewhat different. He was led to this conclusion by finding that potassium produced an effect similar to that of lead in suppressing detonation in an engine. Since potassium is so rapidly oxidized, it must be supposed that no uncombined metal would have time to reach the fuel. When, in the author's variable compression engine, lead vapour was produced by an electric arc and was carried with the air and fuel into the cylinder, it was found to reduce detonation. When thallium was used in the arc the effect was very much greater. Whether in either metal the arc was struck in an atmosphere of air or of nitrogen, the suppression of detonation was equally marked. Since thallium and probably also lead vapour under such conditions would be almost instantly oxidized in air, it could not be doubted that these metals even in the form of oxides could serve to suppress detonation if in a sufficiently divided state. It was observed, furthermore, that the metals which are effective as "anti-knocks" are capable of forming two oxides, and that the temperatures of transitions of these oxides were found to be within the range of temperature which occurs during compression in an engine.

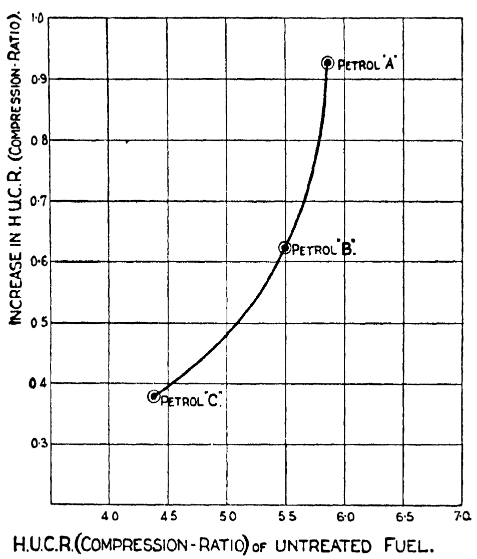

Fig. 24

In Egerton's conception of the processes of combustion he supposes the existence of a chain of intermediate products. By its interference with this chain of reactions the active metallic oxide is believed to work. To account for the effectiveness of so small a concentration, it is suggested that the higher oxide of the metal is reduced to the lower, thereby oxidizing the intermediate organic product which would otherwise give rise to detonation. On next meeting an oxygen molecule the now lower metallic oxide is reconverted to the highest form. By such means the molecule is rejuvenated with enormous frequency and is so enabled to work over and over again.

It seems that the function of the anti-knock is to do quietly and perhaps by a less roundabout route that which in conditions giving rise to detonation takes place with great violence. Normally, with a given fuel, the compression ratio may be raised until the " end " charge reaches such a condition that detonation occurs in some part of it. The effect of the anti-knock is to allow this process to go further in greater or less degree, according to the effectiveness of the anti-knock and the concentration used in the fuel.

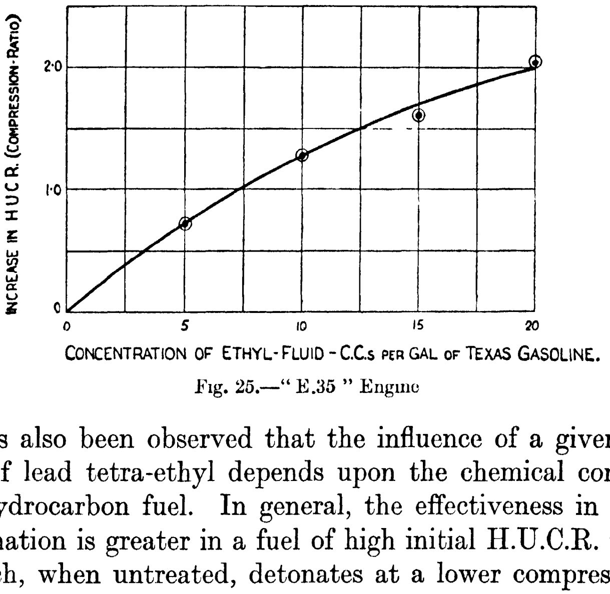

Fig. 25.—" E.35 " Engine

It has also been observed that the influence of a given concentration of lead tetra-ethyl depends upon the chemical composition of the hydrocarbon fuel. In general, the effectiveness in suppressing detonation is greater in a fuel of high initial H.U.C.R. than in a fuel which, when untreated, detonates at a lower compression.

Example of Influence of Fuel on the Effectiveness of an " Anti-knock ".—In fig. 24 is summarized in curve form the results of a number of tests wherein, in each case, 5 c. c. " Ethyl Fluid " (seven-twelfths of which by volume consisted of lead tetra-ethyl) were added to a gallon each of three widely different samples of petrol. From the graph it will be seen that this concentration raised the H.U.C.R. of a very good petrol by 0·93 ratios, of a moderately good one by 0·62 ratios, and of a very poor petrol by only 0·38 ratios.

Examples of Effectiveness of an "Anti-knock" in different Concentrations.—Fig. 25 illustrates the increase of H.U.C.R. due to the presence of " Ethyl Fluid " in various concentrations. It is seen that the steepness of the curve falls off gradually as the concentration is increased. Even up to 20 c. c. however, (at which concentration the compression ratio could be raised by 2 ratios, corresponding to an increase in engine efficiency of 13 per cent) there is no evidence that a further addition would not have been effective in raising the H.U.C.R. The curves in figs. 26 and 27 are of interest as showing how, for two entirely different types of engine (fig. 26 refers to the small sleeve-valve engine " E.5 ", while fig. 27 relates to the large poppet valve " E.35 "), the effectiveness

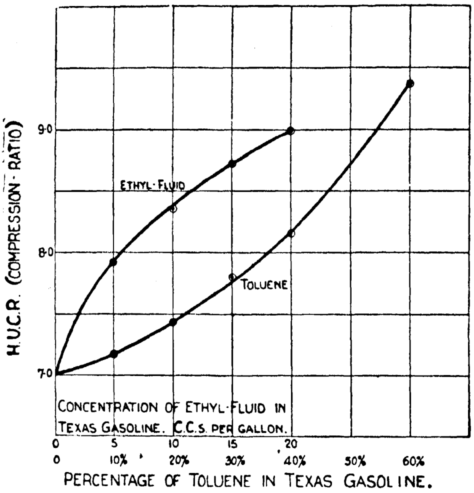

Fig. 26

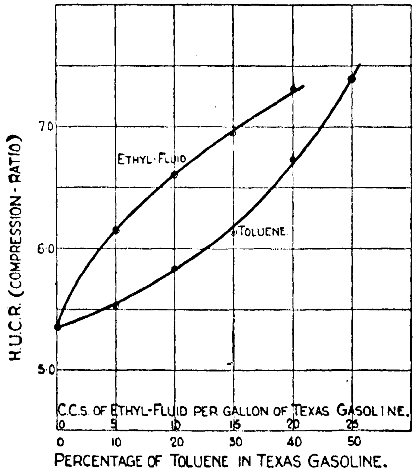

Fig. 27.—" E.35 " Engine

of the metallic " anti-knock " diminishes, while the effectiveness of pure toluene increases, with increasing concentrations of each.

Example of Effectiveness of an "Anti-knock" in different Types of Engine.—In fig. 28 are plotted from the two preceding graphs, curves showing the effectiveness of " Ethyl Fluid " in terms of the percentage of toluene giving the same increase of H.U.C.R. From this graph, since the two curves are on the same scales, the effects of various concentrations are readily comparable for the two very different types and sizes of engine. It is interesting to observe that the curves are so nearly parallel and that the difference in effectiveness is at no point very great, when it is considered that the variations in design factors are almost the widest possible.

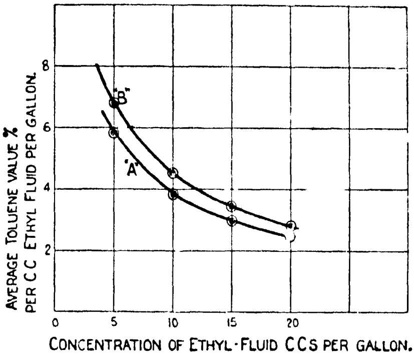

Fig. 28.—Curve " A ", " E.35 " Engine
„ " B ", " E.5 " „

Effectiveness of Tetra-ethyl Lead with Varying Inlet Temperature.—The precise nature of the series of chemical events leading to detonation is still uncertain. If, however, the theory is correct that peroxides play a large part and that the lead molecules are used many times over during the compression stroke and the early stages of combustion, their effectiveness might be expected to be a function of the temperature of the gases during compression and the turbulence of the charge. A rise in air inlet temperature will render the air fuel charge a more nearly homogeneous gaseous mixture and will speed up the chemical reactions and enable the lead to do a greater amount of repetition work in the available time, with the result that its effectiveness is increased. Fig. 29 shows the effect of increasing heat input to the induction pipe at a constant speed on two fuels, one a normal aromatic petrol and the other a petrol containing approximately 1·8 mls. tetra-ethyl lead per gallon. It will be seen that both fuels have the same H.U.C.R. at a heat input of 70 watts; when the heat input is increased to 500 watts, however, the difference in H.U.C.R. is nearly 0·4 ratio in favour of the leaded fuel.

By the same reasoning the greater effectiveness of lead when

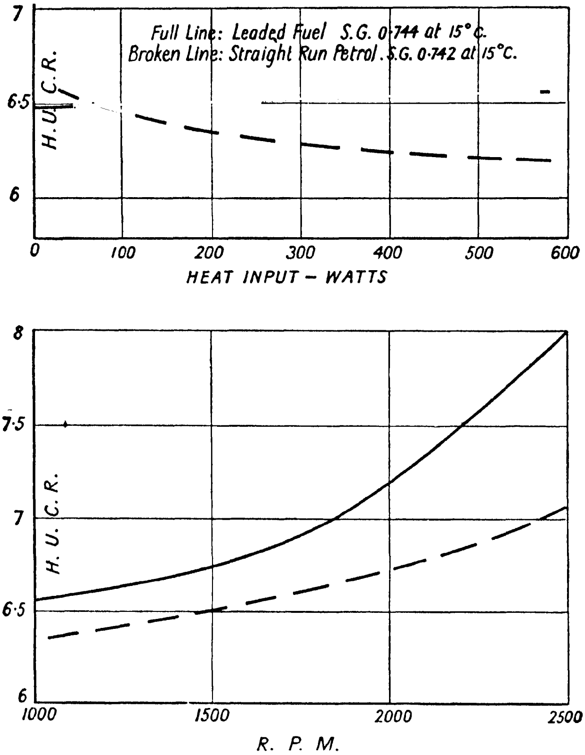

Fig. 29

used with fuels of higher initial H.U.C.R. may be explained (fig. 24). The higher compression temperature at the higher compression ratio encourages the chemical reaction of the lead oxides, enabling them to begin their work earlier and thus giving longer time for the oxidation of the chain compounds which would otherwise cause detonation. Moreover, not only is the time prolonged but the general level of temperature is higher throughout, thus greatly accelerating the oxidation and reduction of the lead oxides.

Effectiveness of Tetra-ethyl Lead in relation to Speed.—In the diagram (fig. 29) the H.U.C.R. of two fuels in relation to speed is also shown. Both fuels have very nearly the same anti-knock value up to 1500 R.P.M., and the same specific gravity. One fuel is a normal straight run petrol, while the other consists of a petrol of rather lower anti-knock value which has been treated by the addition of 1·8 mls. lead tetra-ethyl per gallon. It is seen that the effectiveness of the lead content increases as the speed is raised, so that at 2000 R.P.M. the difference in H.U.C.R. is nearly 1 ratio in favour of the leaded fuel.

The Use of Lead Tetra-ethyl as an Anti-knock.—The effectiveness of lead in small concentrations has made it a most convenient substance to use to bring basic fuels of varying octane rating up to a required standard. Moreover, with the heavy demands for fuels of high knock ratings, ranging from 77 to 100 octane numbers or over, the use of lead has been an invaluable means of adding up to 10–12 octane numbers to fuels of moderate anti-knock quality. In this way fuel which would of itself be below the standard required can be used with a small concentration of lead for purposes which, failing lead, would require a basic fuel of high anti-knock value, a product both expensive to manufacture and more limited in supply.

In the early days the residues left behind in the form of lead compounds and the corrosive effects of these deposits were regarded as a serious deterrent to the use of lead. With the introduction of non-corroding materials and protective coatings for the vulnerable parts, however, the use of lead in concentrations up to a value beyond which, in any case, the return in anti-knock quality is seriously diminishing, no longer gives cause for anxiety, and to-day leaded fuel is as freely and widely employed as fuels not containing lead.

For petrols up to octane number 77, i.e. within the range required for commercial vehicles and the private motor-car, lead is not in general use, as there is an adequate supply of cracked and synthetic fuel of sufficiently high anti-knock value.

The commercial air lines use fuels of 87 octane number, while aero-engines for military aircraft are now designed to employ 100 octane fuel. In the case of fuel for the commercial air lines, lead tetra-ethyl to a maximum concentration of approximately 3 mls. per gallon may be used, while for military purposes a lead content up to 4 mls. per gallon is allowed. As a rough estimate the addition of the first ml. increases the octane number by 6, the second by 3, the third by 2, and the fourth by 1.

Fuel Sensitivity.—The production of high-grade fuels by "cracking" processes has brought with it its own special difficulties and problems. One of these is the difference in behaviour of these fuels, when compared with straight run distillates, in their tendency to detonate in an engine in service.

It is found that cracked fuels are in many instances more sensitive to changes in engine conditions than straight run fuels and that the order of their knock-rating may in some instances be entirely altered by such changes.

In these circumstances the use of fuels of which the octane rating has been determined by the standard method is attended by uncertainty in regard to their detonation behaviour in a road-vehicle engine in which there may be variations in mixture ratio, spark phasing between cylinders, or some other condition to which such fuels are sensitive.

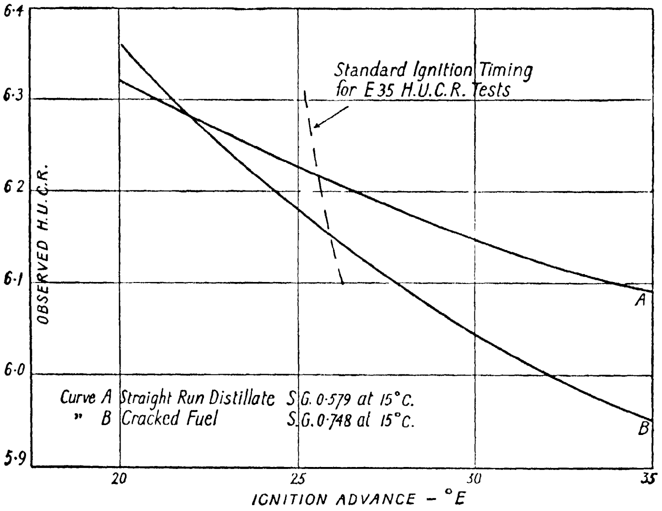

Fig. 30

As one example of fuel sensitivity, two curves of H.U.C.R. obtained on the "E.35" variable compression engine over a wide range of spark advance are given, one with a straight distillate petrol of H.U.C.R. 6·22 and the other a cracked fuel of H.U.C.R. 6·15 (see fig. 30).

It is seen that the effect of change in ignition advance on H.U.C.R. is markedly different in the two cases, the cracked spirit "B" being nearly twice as sensitive as fuel "A". It is easy to see that an error in spark phasing in an engine running on the cracked fuel will have a much greater effect on detonation than if the engine were operating on the straight-run distillate.

The question of fuel sensitivity, particularly in regard to the behaviour of modern fuels, is one about which comparatively little is known at the present time.

Range of Burning.—So long as a fuel contains no free hydrogen, the available range of burning differs but little between the different volatile liquid fuels and is, compared with illuminating or other gaseous fuels, very narrow indeed. It is clear that so far as efficiency is concerned, it is only the range of burning on the weak side that need be taken into consideration; the range of burning on the rich side, that is up to the point at which the mixture fails to ignite owing to over-richness, is of comparatively little practical interest. It has already been shown that any range on the weak side is of the utmost importance, for so long as the fuel will burn completely, the weaker the mixture, the lower the flame temperature, and therefore the higher the efficiency. If combustion were complete the flame temperature would diminish very nearly in proportion to the mixture strength. Unfortunately, incomplete and delayed combustion become apparent so soon as the mixture strength is reduced by more than about 15 per cent below that giving complete combustion; from 12 per cent to 18 per cent weak, the loss by delayed and incomplete combustion and the gain due to the lower flame temperature, just about balance each other, while beyond about 18 per cent weak the net efficiency begins to fall away rapidly, and the rate of burning becomes so slow that it continues throughout the exhaust stroke and so ignites the fresh charge on its entry to the cylinder, causing the familiar back-fire into the induction system. This can, however, be obviated to a limited extent by advancing the time of ignition, and the range on the weak side can be extended slightly in consequence, but the angle of ignition advance required, efficiently to burn a mixture only 20 per cent weak, is so excessive as, in practice, to be almost impracticable. In fig. 31 is shown a typical curve showing the relation between thermal efficiency, mean pressure and mixture strength with fixed ignition timing. The mixture strength is plotted in terms of percentage excess of fuel or air. In fig. 32 is shown a similar curve, but with the ignition advanced a further 30° between 10 per cent and 20 per cent weak; in both cases the mean pressure and efficiency were obtained by direct measurement, and represent the mean of a large number of tests on various fuels. The point at which combustion is complete, that is, at which there is no excess either of fuel or air, is shown by a vertical line, to the right of which

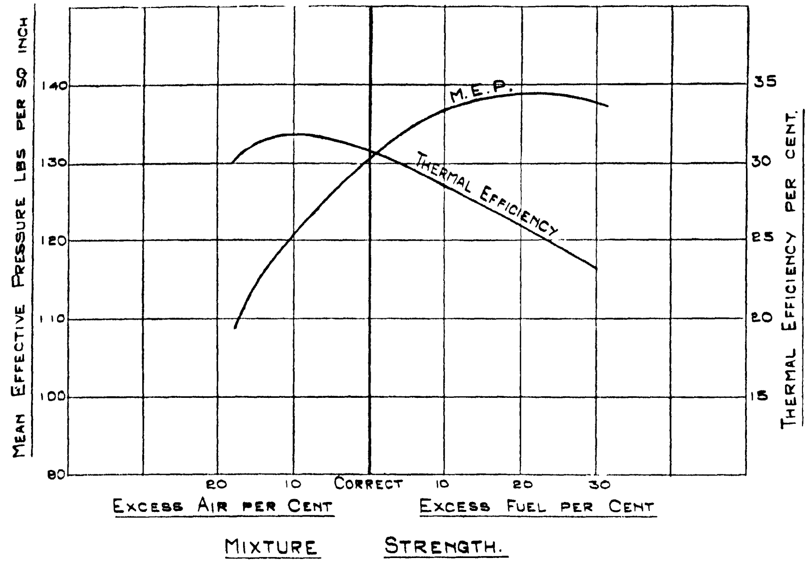

Fig. 31.—Indicated Mean Pressure and Thermal Efficiency at different Mixture Strengths with fixed Ignition Timing. Fuel, Petrol

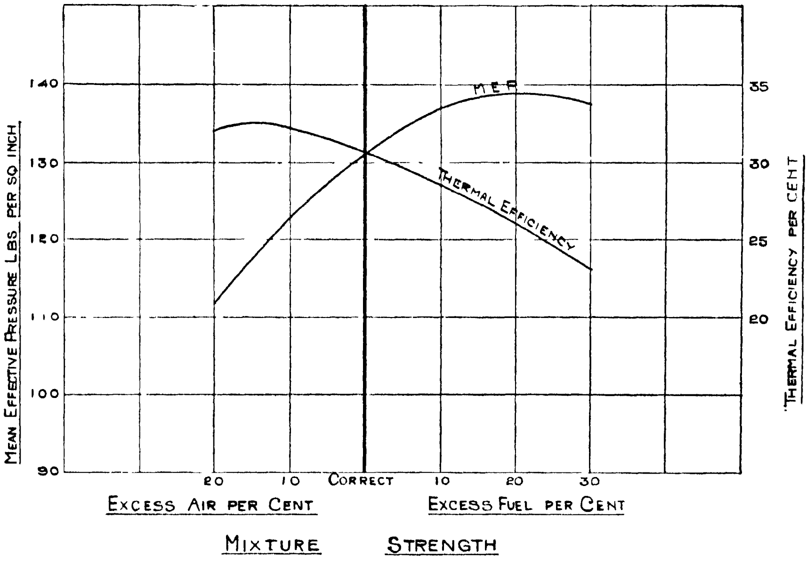

Fig 32.—Indicated Mean Pressure and Thermal Efficiency at different Mixture Strengths with Ignition Timing adjusted for each Change in Mixture Strength. Fuel, Petrol

the mixture is over rich and to the left it is weak. In fig. 33 is shown a similar curve taken with alcohol. It will be observed that while, in each case, the M.E.P. increases as the mixture is enriched beyond the point of complete combustion, it is only in the case of alcohol that there is any increase beyond 20 per cent excess of fuel. The increase in mean pressure with excess fuel depends upon the inter-relation of a number of factors; on the one hand are the increase in specific volume and the increase in volumetric efficiency due to the latent heat of the fuel, both of which tend to increase the power output with further increase of mixture strength. Against these must be offset the higher specific heat of the products of combustion, when the fuel is only partially burnt. In the case of petrol and benzol the increase is small, because the latent heat is low and the several factors more or less balance one another. The latent heat of benzol is higher than petrol, and moreover more use can be made of it, because the fuel is homogeneous, but, on the other hand, the change in specific volume is smaller. In the case of alcohol, the latent heat is very much higher and the change in specific volume also is greater, consequently with this fuel the mean pressure increases as the mixture is enriched to a far greater extent than with either of the other two.

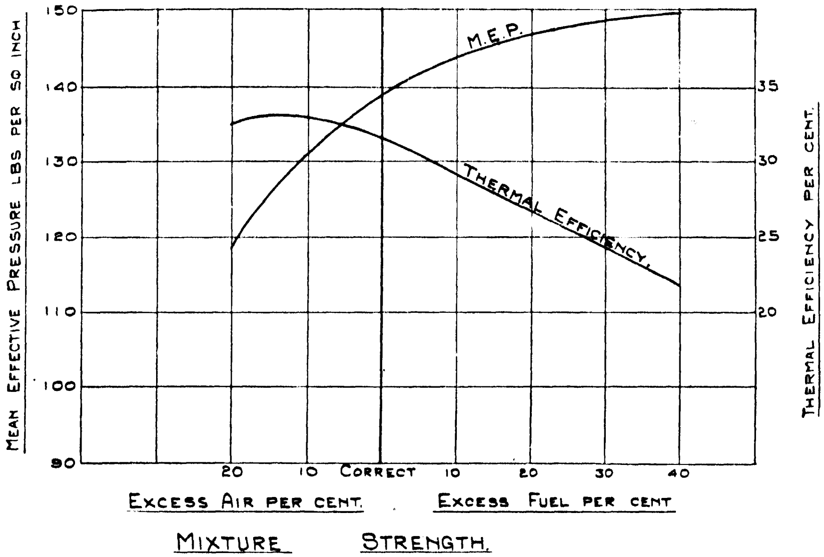

Fig. 33.—Indicated Mean Pressure and Thermal Efficiency at different Mixture Strengths with Ignition Timing adjusted for each Change in Mixture Strength. Fuel, Alcohol

The range of burning on the weak side is substantially the same in all three cases and indeed in the case of all known volatile liquid fuels.

It will be observed that the efficiency is at a maximum when the excess of air is between 10 and 18 per cent; with an excess of 20 per cent the process of combustion becomes so slow that the running of the engine is unstable, and both misfiring and back-firing through the inlet valve are liable to occur. The tests on which these results are based were all carried out on single-cylinder engines. In the case of a multi-cylinder engine, however well designed the distribution system may be, it is practically impossible to maintain the mixture strength to all cylinders uniform to within closer limits than about 10 to 20 per cent. If the mean strength supplied by the carburettor is, say, 15 per cent weak, it follows that one or more cylinders will receive mixture as much as 20 to 25 per cent weak which will result in misfiring and back-firing, through the carburettor. To allow therefore for inequalities in distribution, it is necessary, in practice, even in the very best examples to supply a mixture containing not more than about 10 per cent excess of air. With such a mixture the supply to some cylinders will be about that giving complete combustion to others about 20 per cent weak, and the maximum efficiency obtainable will be about 1·5 per cent lower than could be obtained from a single-cylinder engine. It will be seen, therefore, that in order to legislate for the weakest cylinder and prevent back-firing, one or more of the remainder must receive a mixture which is too rich for maximum economy, and the indicated thermal efficiency of a multi-cylinder engine will, on this account, always be lower than that of a single cylinder. Further, the greater the number of cylinders fed from any one source of supply, the lower the efficiency.

To sum up, (1) the available range of mixture strength on the weak side, with all volatile hydrocarbon fuels, is very narrow, far narrower than with most of the gaseous fuels.

(2) Owing to the narrow range of available mixture strength and to the inevitable inequalities in distribution, multi-cylinder engines cannot run with the most economical mixture strength, hence their fuel consumption per H.P. hour must always be slightly higher than that of a single cylinder—how much higher, depends upon the number of cylinders supplied by any one carburettor and, of course, on the efficiency of the distribution system.

(3) With all fuels, slightly more power is developed with an over-rich mixture as compared with the mixture giving complete combustion; with alcohol the increase in power is very marked, and amounts to nearly 10 per cent with very rich mixtures.

(4) In the case of single-cylinder engines running on petrol or benzol, maximum economy is obtained at about 92 to 94 per cent of full power; in the case of multi-cylinder engines, for the reasons given above, the maximum economy on the same fuels is slightly lower, but is obtained at about 96 to 97 per cent of full power.

The Temperatures of the Cycle. From the data and information now available, it is possible to deduce with a fair degree of accuracy, the temperature changes throughout the cycle when liquid hydrocarbon fuels are used.

The first temperature which has to be determined is that at the end of the suction stroke, for from this all the other cycle temperatures can be obtained. It is also of direct importance as determining the volumetric efficiency of the engine.

The temperature at this point is influenced by the final exhaust temperature, owing to the admixture of the fresh charge with the residual exhaust products in the clearance space. It therefore should strictly be evaluated by a "hit-and-miss" type of calculation, by which a residual exhaust temperature is assumed, and the cycle temperatures worked out on that assumption. The assumed exhaust temperature is then modified till it agrees with that which is obtained by calculating round the cycle.

Fortunately, however, it requires a large change in exhaust temperature appreciably to alter the suction-temperature, so that the assumed and calculated temperatures need not check very closely.

It is, in the author's opinion, always preferable to take a concrete example; we will therefore consider the case of a cylinder with a swept volume of 80 cub. in. running at 2000 R.P.M. Let the compression ratio be 5 : 1, making the total cylinder volume 100 cub. in., and let the following conditions be assumed :—

Mean jacket temperature	140° F.
Heat input to charge external to cylinder (by carburettor heating, &c.)	0·05 B.Th.U. per cycle.
Absolute pressure in cylinder at end of suction stroke	14·0 lb./in.
Absolute pressure in cylinder at end of exhaust stroke	14·7 „
Mean temperature of outside air	60° F.

These are conditions which may be taken to represent average practice.

The fuel is assumed to be an average high-class petrol of composition say 50 per cent paraffins, 35 per cent naphthenes, and 15 per cent aromatics, while the case of ethyl alcohol and benzene (benzol) are also dealt with.

The relevant properties of these fuels are :

Fuel.	S.G.	Effective Cal. Val. for B.Th.U./lb.	Latent Heat of Evaporation, B.Th.U./lb.	Boiling Point.
Petrol	0·740	19,000	135	160°-400° F.
Benzene	0·884	17,460	172	176° F.
Ethyl alcohol ...	0·798	11,840	397	172° F.

Fuel.	Change in Sp. Vol. after Combustion (Correct Mixtures).	Complete Combustion Mixture (by Weight).	Energy liberated per Standard cub. in. of Mixture ft. lb. exclusive of change of Specific Volume.
	per cent.		
Petrol	+5·0	14·3/1	46·2
Benzene	+1·3	13·2/1	46·9
Alcohol...	+6·5	8·95/1	44·5

Consider first the conditions when running on petrol, starting with the commencement of the suction stroke. The cylinder then contains 20 cub. in. of hot exhaust products at atmospheric pressure. These will be at a temperature of about 2100° F. absolute, as will be shown later. It is desirable in order to arrive at their heat capacity relative to that of the entering charge to reduce them to terms of normal temperature and pressure. The volume of the residual exhaust products under these conditions will therefore be $20 \times \frac{491}{2100} = 4{\cdot}68$ cub. in.

The incoming charge consists of air at a temperature of about 60° F. and a small proportion of fuel, entirely or partly vaporized. The air/fuel ratio for complete combustion will, in the case of petrol, be 14·3/1. The latent heat of evaporation of 1 lb. of petrol is 135 B.Th.U.s. This is supplied by 14·3 lb. of air of which the specific heat at constant pressure is 0·237; the drop in temperature of this air will be $\frac{135}{14{\cdot}3 \times 0{\cdot}237} = 40°$ F.

For the mass of air in question 1° F. change in temperature is produced by ·00067 B.Th.U., so that the latent heat of evaporation will take up $40 \times \cdot 00067 = \cdot 0268$ B.Th.U. From the cylinder walls at a temperature of about 140° F., and from the still hotter valve and piston surfaces, &c., the charge picks up about 0·0005 B.Th.U. per cub. in., or in this case 0·04 B.Th.U. per cycle. The net gain of sensible heat to the charge is $0{\cdot}04 + 0{\cdot}05 - 0{\cdot}0268 = 0{\cdot}0632$ B.Th.U. per cycle, which will raise the temperature by about 95° F. to 155° F., assuming that all the fuel is completely evaporated. If the direct heating does not complete evaporation, mixture with the hot residual gases will certainly do so unless the mechanical pulverization of the fuel in the carburettor is very inadequate. It follows therefore that

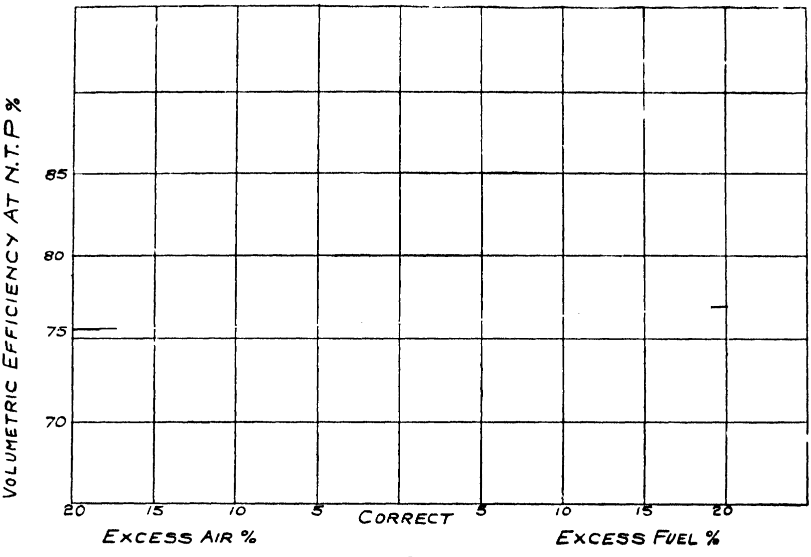

Fig 34.—Observed Volumetric Efficiency with different Mixture Strengths at a Compression Ratio of 5·0 : 1. Fuel, Petrol

the final suction temperature will be the same at whatever stage the latent heat of evaporation is extracted, provided only that evaporation is complete before compression starts.

We now have 80 cub. in. of fresh charge at 155° F. or 614° F. abs. and 14·0 lb./in. absolute pressure. Reduced to N.T.P. this becomes $80 \times \frac{14{\cdot}0}{14{\cdot}7} \times \frac{491}{614} = 60{\cdot}9$ cub. in. This gives a volumetric efficiency of $\frac{60{\cdot}9}{80} = 76{\cdot}2$ per cent, a figure which agrees very closely with that found in, for example, the author's single-cylinder research engine under similar temperature conditions, as shown in fig. 25, which indicates the observed volumetric efficiency at different mixture strengths when running with the heat input to the carburettor specified above, and at a piston speed of 2000 ft. per minute.

The volume of the residual exhaust gases at N.T.P. was shown to be 4·68 cub. in., so that the total standard volume of the combined mixture $= 60{\cdot}9 + 4{\cdot}68 = 65{\cdot}58$ cub. in. As this actually fills a volume of 100 cub. in. at a pressure of 14·0 lb./in. its temperature must be $\frac{100 \times 14{\cdot}0}{14{\cdot}7 \times 65{\cdot}58} \times 491 = 717°$ F. absolute or 258° F.

This figure may be taken as the final suction temperature to within about ±10° F. in the case considered. The chief possibilities of error lie in

(1) The temperature of the exhaust products. Owing, however, to the fact that as their temperature increases their mass correspondingly decreases, a large error here has but little influence on the suction temperature.

(2) The heat picked up from the cylinder walls, &c. This is certainly rather a doubtful figure, but the author has arrived at it by running an engine with varying jacket temperatures, and adding known quantities of heat by means of an electrical resistance in the induction pipe until the volumetric efficiency, and therefore the suction temperature, was constant. By measuring the difference in external heat necessary to do this with varying jacket temperatures, it was possible to estimate the total amount of heat taken up from this source.

(3) In most cases the amount of pre-heating of the charge before its entry to the cylinder is very uncertain, since in practice the heat is generally supplied either from the exhaust or from the circulating water. For experimental work the author prefers to supply this heat electrically so that it can readily and accurately be measured.

If benzene be used instead of petrol, we find that, owing both to its higher latent heat and to the greater proportion of fuel needed to combine with the air, the final suction temperature is lower, namely, about 235° F., while the volumetric efficiency is correspondingly higher, about 78·5 per cent. The residual exhaust temperature may be taken, as with petrol, as 2100° F. absolute.

In the third case, that of ethyl alcohol, the extremely high latent heat and the large proportion of fuel in the mixture produce much greater cooling effect, with the result that, arguing on the above premises, the final suction temperature, even after admixture with the residual exhaust products, will only be 67° F.; the corresponding volumetric efficiency should therefore be 104·3 per cent.

At such a temperature, however, and in the short time available, the alcohol would not be evaporated completely by the end of the suction stroke, so that such a calculation is not applicable in this instance. Experiment has shown that the minimum temperature at which evaporation is completed before compression, is in the neighbourhood of 150° F., and it will be well to consider the case in which sufficient pre-heating is supplied to effect this. In this case the volumetric efficiency works out at 90 per cent, and 37 per cent of the fuel remains to be evaporated by means of additional external heating to the extent of ·37 × 0·143 = ·053 B.Th.U. per cycle.

In practice it is found that the volumetric efficiency on alcohol is considerably lower than 90 per cent, namely, 82-83 per cent as compared with 76 per cent in the case of petrol. This shows that under normal working conditions only a comparatively small proportion of the alcohol is actually evaporated by the end of the suction stroke. It is mainly for this reason that efficient pulverization, which accelerates the transfer of heat from the air to the fuel particles, is so particularly desirable when using alcohol.

The above conclusions may be summarized as follows. Under the conditions quoted above—and they may be taken as average working conditions, except perhaps in the case of alcohol—the final suction temperature and the volumetric efficiency will be :

Fuel.	Suction Temperature.	Volumetric Efficiency.
	° F.	per cent.
Petrol	258	76·2
Benzene...	235	78·5
Alcohol	150	90·0

With a 20 per cent weak mixture the figures become :

Fuel.	Suction Temperature.	Volumetric Efficiency.
Petrol	263	75·5
Benzene...	242	77·6
Alcohol	150	89·6

In the case of alcohol extra heat to the amount of 0·033 B.Th.U.

per cycle has to be supplied, and 0·029 B.Th.U. with the weak mixture, if evaporation is to be complete.

It should be noted that the volumetric efficiency refers to the volume of air and fuel vapour, not air alone.

Compression Temperature.—During the compression stroke the fuel and air are compressed nearly adiabatically to one-fifth of their original volume. At the commencement of compression the mixture, in the case of petrol or benzene, is slightly below the mean surface temperature of the cylinder, so at first they will absorb heat, but will lose heat later in the stroke. For these fuels the mean value of the exponent during compression for such an engine running at 2000 R.P.M. may be taken as 1·35. With alcohol the conditions are not quite the same, for

(1) The final suction temperature is substantially below the mean wall temperature, so that the mixture is appreciably heated during the earlier part of the stroke.

(2) As the alcohol forms an appreciable fraction of the charge (11 per cent by weight), it will decrease the mean exponent γ for the mixture, since γ for alcohol vapour is only 1·13 as against 1·4 for air.

(3) A considerable portion of the fuel is liquid at the end of the suction stroke, unless extra induction heat is supplied, and some of the heat of compression is applied to its evaporation.

We will ignore for the moment the last factor, and take γ for a " correct " alcohol mixture as being about 1·33.

The absolute temperature at the conclusion of the compression stroke in the case of petrol or benzene may be arrived at by multiplying the final absolute suction temperature by $5^{(1\cdot35-1)} = 1\cdot755$; This gives :

Petrol	correct mixture	1258° F. abs.
	20 per cent weak mixture	1267° F. abs.
Benzol	correct mixture	1220° F. abs.
	20 per cent weak mixture	1232° F. abs.

The compression pressure in each case will be $14\cdot0 \times 5^{(1\cdot35)} = 123$ lb. per square inch absolute or 108·3 lb. per square inch gauge.

With the alcohol mixtures, the final suction temperatures are the same both for the normal and the weak mixtures, so that the compression temperature, uncorrected for the latent heat of the surplus liquid, are also the same, being

$$609 \times 5^{(1\cdot33-1)} = 1035° \text{ F. abs.}$$

With the correct mixture there is, as previously shown, a heat deficit of 0·053 B.Th.U. per cycle, this amount being necessary for the evaporation of the alcohol which remains in a liquid state at the end of the suction stroke. This causes a drop of $0{\cdot}053 \times 1270 = 67°$ F., making the actual compression temperature 968° F. abs. or 475° F.

With the weak mixture, the heat deficit is only 0·029 B.Th.U. per cycle, which raises the compression temperature to 998° F. abs. or 505° F.

The uncorrected compression pressure is $14{\cdot}0 \times 5^{(1{\cdot}33)} = 119$ lb. per square inch absolute. The cooling due to evaporation reduces this to 111 lb. per square inch abs. for the normal mixture, and 114·7 lb. per square inch abs. for the weak mixture.

Combustion Temperatures.—Since combustion takes place at constant volume, the whole of the chemical energy stored in the mixture, with the exception of that lost to the walls of the combustion chamber, will be devoted to increasing its internal energy. It is well known, however, that the obvious method of finding the temperature rise, that of dividing the heat output by the specific heat at normal temperatures, gives results that are far too high, in fact practically double those that are actually attained. This discrepancy is due to the three following influences :—

(1) The specific heat of the gases, of which the working fluid is composed, increases with increase of temperature.

(2) At high temperatures the products of combustion, CO_2 and water, dissociate with absorption of heat into carbon monoxide and oxygen, and hydrogen and oxygen respectively, according to the formulæ

$$2CO_2 \leftrightarrows 2CO + O_2$$
$$2H_2O \leftrightarrows 2H_2 + O_2.$$

(3) Heat is lost by radiation and conduction to the walls of the combustion chamber.

The specific heat of the gases, of which the working fluid is composed, and the amount of dissociation at any temperature, have been found by direct experiment, while the loss to the jackets can only be estimated from actual engine experiments. The amount of energy taken up by the gases as sensible heat can readily be calculated, since the specific heat of a gas at constant volume is unaffected by changes of pressure or by admixture with other gases. Unfortunately the degree of dissociation is influenced by alterations in pressure and, in cases such as the present, where two kinds of dissociation have a common product (oxygen in this case), by the relative proportions of the gases present. This renders the actual calculation of the energy absorbed in this manner a somewhat complex and extremely wearisome process; it has, however, been dealt with very fully by Tizard and Pye, so that the results alone will be given. The most convenient method is to construct a curve showing the total internal energy of the mixture, at any temperature, plotted against the temperature. Since the amount of internal energy at any stage in the cycle can easily be found, the corresponding temperatures can at once be read off.

From the work of Tizard and Pye[1] which is based on the experimental data of Pier and Bjerrum[2] the curve shown in fig. 35 has been constructed, which also gives a graphical construction developed by one of the author's assistants, Mr. J. F. Alcock, whereby the temperatures at the beginning and end of the expansion stroke can be read off with a fair degree of accuracy. Owing to the influence, previously mentioned, of pressure and charge proportions on dissociation the construction of such a curve to meet a reasonably wide range of conditions renders it necessary to compromise to some extent and to take mean values.

Thus, the energy curve shown applies strictly only to a benzene-air complete-combustion mixture at a compression ratio of 5 to 1. However, the effect of a change in the compression ratio is almost negligible within the limits used on the constant-volume cycle; as is the effect, also, of replacing benzene by any other hydrocarbon fuel. If used for a fuel such as alcohol or ether, however, the error becomes appreciable, owing to the different specific heat of the products of combustion. Strictly speaking, it does not apply to mixtures which are either weaker or richer than the chemically correct mixture, although here again the divergence is very small within the range available and with a homogeneous mixture.

The internal energy curve is plotted in terms of foot-pounds per standard cubic inch on a vertical scale, against the temperature on a horizontal scale. The other full-line curve shows the energy present as heat, so that the difference between the two curves shows the chemical energy stored in the products of dissociation. Zero energy is taken at 100° C. (212° F.), as being an average temperature at the beginning of compression. Variations in this temperature, being relatively small, will have but little influence on the combustion and exhaust temperatures. The explanation given below of the use of the diagram is supplemented

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[1] *Automobile Engineer*, February and March 1921.

[2] *Zeitschrift fur physikalische Chemie*, 1912.

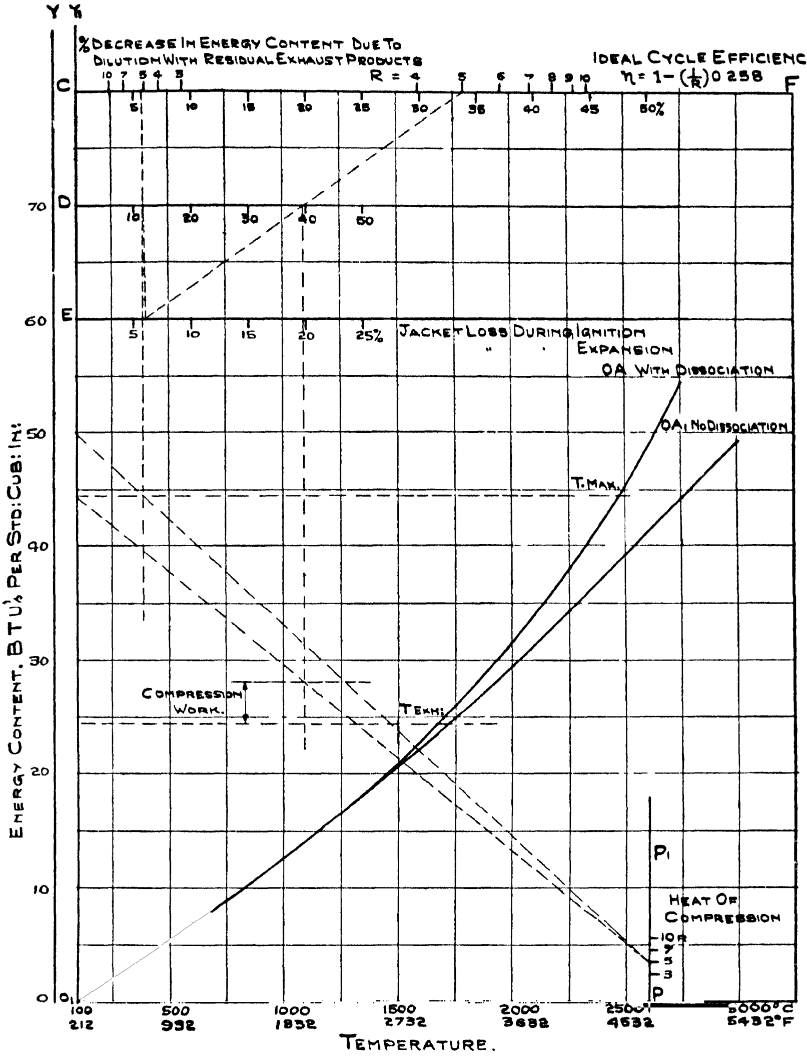


Fig 35. —Internal Energy Diagram for working Fluid of Internal-Combustion Engine running on Volatile Hydrocarbon Fuels

by an example worked out for the following data, the construction lines of the example being shown dotted in the figure.

Compression ratio, R = 5 to 1
Energy content = 46·2 ft. lb. per cub. in.
Heat loss during combustion = 6 per cent } The actual value of these losses
Heat loss during expansion = 6 per cent } will be considered in Chap. III.

There are three factors in an actual engine which modify the *temperature attained by the combustion of a mixture of any given energy content. They are the*

(1) Heat put into the mixture by compression.

(2) Loss due to cooling by the walls of the combustion chamber during combustion.

(3) Effective weakening of the mixture due to dilution with the residual exhaust products.

Factor (1) is allowed for by laying off the heats of compression for various ratios by the marks "R – 5," &c., on the line PP_1 near the bottom of the diagram. The energy content is then marked off above this on the vertical line O_1Y_1 representing the 100° C. (212° F.) starting-point. In the example, the 46·2 ft. lb. energy content is laid off above the 3·6 ft. lb. of compression, making a total of 49·8 ft. lb., this being the gross energy content from which the losses due to factors (2) and (3) must be deducted. These are provided for in the following manner: –

On the horizontal scale C is marked the effective energy loss due to dilution with residual exhaust, assumed to be at 2100° F. abs. (= 1165° C. abs.). Scale E shows the percentage loss due to cooling during combustion. This is laid off at any figure which previous experience shows as probable for the type of combustion chamber in question, namely, 6 per cent in the example. A line is then drawn between these two points, and the point of intersection of the line so drawn with the scale D gives the total percentage loss due to these two causes, corresponding to 11·5 per cent in the example chosen.

To transfer this to the diagram, a line is dropped vertically from the above intersection point. Another line is drawn from the point on the line O_1Y_1, giving the gross British thermal units per cubic inch, to the suitable compression point on the line PP_1, and representing 100 per cent on scale D; or 49·8 ft. lb. per cub. in. in the example. From the intersection of the above two lines, a horizontal line is run to the energy scale on one side, and to the energy curve on the other. The point on the energy scale shows the net energy available for expansion, or 44·5 ft. lb. per cub. in. in the example. From the energy-temperature curve the actual flame temperature can be read off, this is 2475° C. (4487° F.) in the example.

The drop in temperature during the expansion stroke depends on the two factors of (*a*) external work done, and (*b*) heat loss to the walls. The net power output is given as a percentage of the *heat content of the mixture on scale F, the formula used being* $n = 1 - \left(\frac{1}{r}\right)^{0.258}$. *0·258 is the mean value for the exponent, and* covers all dissociation and similar effects, but not wall losses. The wall loss during expansion is laid off on scale E. A line drawn between these points gives their sum on scale D as before. A perpendicular from this point is dropped to meet a line from the net-energy point on O_1Y_1 to the suitable-compression point on the line PP_1. As the gross work done during expansion is the sum of the net work mentioned above and the compression work, this latter amount, which is 3·6 ft. lb. in the example, must be laid off below the above intersection point, to find the energy content at the end of expansion, which is 24·5 ft. lb. in the example. The corresponding final temperature, 1675° C. (3047° F.) in the example, can then be read off from the energy curve.

Taking an actual example from experiments on the author's variable-compression engine, with a correct mixture of an energy content of 46·2 ft. lb. per cub. in. and a compression ratio of 5 to 1, the actual maximum flame temperature, as obtained from the above diagram in fig. 17, allowing for the additional heat of compression, the wall loss during combustion and the dilution by residual exhaust products, will be 2475° C. (4487° F.), corresponding to an energy content of 44·5 ft. lb. per standard cub. in. At a ratio of 5 to 1, the observed indicated thermal efficiency is 31 per cent; of this, 5 per cent is due to the change in specific volume of the mixture, so that the heat drop is $46{\cdot}2 \times 31 \times 100/105 = 13{\cdot}6$ ft. lb. per cub. in. Add to this the 3·6 ft. lb. of compression work restored during expansion, and the 6 per cent of 46·2 or 2·8 ft. lb. of wall loss during expansion; the total heat-drop during expansion then becomes 20·0 ft. lb. per cub. in., leaving a final energy content of 24·5 ft. lb. per cub. in., which, it will be observed, coincides with the figure found in the example under the same conditions. The corresponding final temperature is 1675° C. (3047° F.)

While affecting the final temperature directly, it should be observed that the loss of heat during expansion has only a slight influence on the actual efficiency; this has been ignored in the construction, because much of it is lost late in the expansion stroke, where its value is less. Another slight error allowed to remain in the construction, for the sake of simplicity, is that a percentage of the *net* heat available during combustion is deducted for the jacket loss during expansion ; whereas this is given as a proportion of the *total* heat available in the fuel. The error due to this cause is, however, very small, being in the case considered $2{\cdot}8\ [(46{\cdot}2 - 44{\cdot}5) \div 44{\cdot}5] = 0{\cdot}11$ ft. lb. per cub. in., and can be ignored safely.

At the end of the expansion stroke the temperature of the gases is about 3050° F. or 3509° F. absolute, as has been seen, and they are at an absolute pressure of about 70 lb. per square inch. When the exhaust valve opens these gases will expand rapidly down to atmospheric pressure, and their temperature will fall in the ratio $\left(\frac{14{\cdot}7}{70}\right)^{\frac{\gamma-1}{\gamma}}$, where γ is approximately 1·30. This brings their temperature down to 2450° F. abs., and the loss of heat during the exhaust stroke further reduces their temperature to about 2100° F. abs.

A similar calculation in the case of an ethyl-alcohol mixture makes the residual temperature about 1950° F. abs.

When the temperature at any point in the cycle has been found, the corresponding pressure is, of course, easily obtained. Thus, in the case of the petrol given above, the compression pressure is 123 lb. per square inch abs. and the temperature 1258° F. abs. As the temperature after ignition is 4487° F., or 4946° F. abs., and the ratio of the specific volumes before and after combustion 1·05, the pressure at the commencement of expansion is therefore $123 \times 1{\cdot}05 \times \frac{4946}{1258} = 508{\cdot}5$ lb. per square inch abs. or 493·8 lb. per square inch gauge.

At the end of the expansion stroke the volume has increased five times, and the temperature has dropped to 3509° F. abs. The pressure is therefore $\frac{123}{5} \times 1{\cdot}05 \times \frac{3509}{1258} = 72{\cdot}2$ lb. per square inch abs., or 57·5 lb. per square inch gauge.

The " explosion " pressure thus calculated is somewhat higher than that actually found in practice due to the finite speed of burning of the mixture, which causes a slight rounding of the peak of the diagram, though not enough appreciably to affect the thermal efficiency.

With benzene, the temperatures at the beginning and end of expansion will be 2470° C. (4480° F.) and 1670° C. (3040° F.) respectively, the corresponding pressures being 505 lb. per square inch abs. (490·3 lb. gauge) and 71·8 lb. per square inch abs. (57·1 lb. gauge) respectively.

From the above considerations we are able to construct an indicator diagram giving the pressures and temperatures throughout the cycle under normal working conditions, such as would apply in the case of a small but efficient cylinder of the size quoted, and when running at a speed of 2000 R.P.M. The influence of cylinder size on the performance will be considered later in Chapter III, when it will be seen that it does not play any very important part; neither, indeed, within reasonable limits does the actual speed of revolution, provided always that it be fairly high, i.e. 1000 R.P.M. or over. At speeds below about 1000 R.P.M. and with a cylinder of the size under consideration, the loss of heat to the cylinder walls will begin to make itself felt, and will influence the performance appreciably.

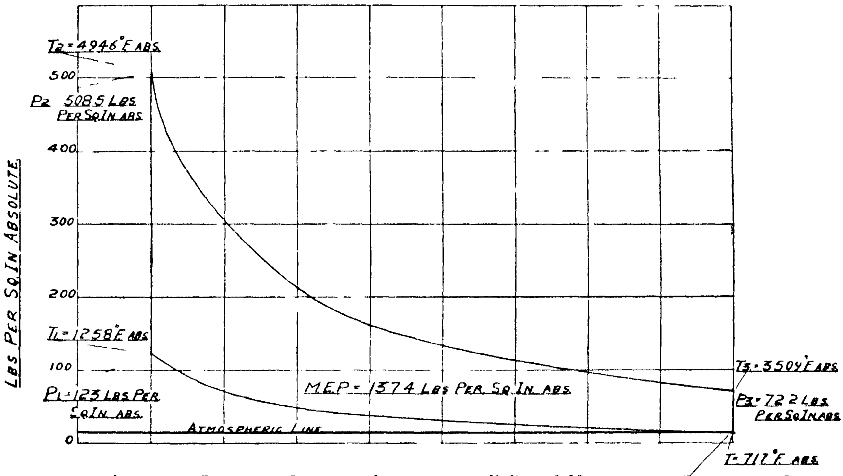


Fig. 36.—Indicator Diagram for "Correct" Petrol Mixture at Compression Ratio 5 : 1

The indicator diagram shown in fig. 36 applies therefore strictly only to the conditions quoted above, though it would differ but little over a wide range of size or speed provided always that the combustion chamber were of an efficient form and that there is ample turbulence to spread combustion rapidly throughout the whole mass of the working fluid.

Once the thermal efficiency, the volumetric efficiency, and the total internal energy of the fuel-air mixture have been determined, it is possible to arrive also at the mean effective pressure and therefore the power output, by multiplying together the thermal efficiency, the total internal energy (in terms of ft. lb. per cubic inch × 12), and the volumetric efficiency.

In the case under consideration the thermal efficiency with a "correct" mixture, i.e. a mixture giving complete combustion without excess either of fuel or air, is 31 per cent. With a 20 per cent weak mixture it should be considerably higher on account of the lower maximum flame temperature, but this, in practice, is offset by the very much slower rate of burning, with the result that it will be increased only to about 32·5 per cent.

The energy liberated by the combustion of 1 standard cubic inch of petrol-air mixture was given as 46·2 ft. lb., but in so far as the available energy from the point of view of power output is concerned, we must multiply this by the change in specific volume, since, if the products of combustion occupy at the same temperature a greater volume than before, the pressures will be higher, though the temperatures remain unaltered. This factor has been taken into account in determining the maximum and terminal pressures, and it, of course, applies also to the mean effective pressure, which will be increased or decreased according as the specific volume increases or decreases; here petrol scores a marked advantage over benzene (benzol), and alcohol over either. It is convenient to use the term "total internal energy of the mixture" (as distinct from the heat energy liberated by combustion) to include the change in specific volume, i.e. the heat energy liberated, multiplied by the specific volume for a "correct" mixture. The total internal energy for the three fuels under consideration then becomes:

Fuel.	Change in Specific Volume.	Heat Energy liberated per Standard cub. in. ft. lb.	Total Internal Energy per Standard cub. in. ft. lb.
	per cent.		
Petrol	+5·0	46·2	48·5
Benzene	+1·3	46·9	47·5
Alcohol	+6·5	44·5	47·4

With petrol and a correct mixture in the example chosen, the thermal efficiency is 31 per cent, the internal energy 48·5 ft. lb. per standard cubic inch, and the volumetric efficiency 76·2 per cent. The indicated mean effective pressure will therefore be

$$\cdot 31 \times (48\cdot 5 \times 12) \times \cdot 762 = 137\cdot 4 \text{ lb. per square inch.}$$

In the case of benzene the same calculation will give

$$\cdot 31 \times (47\cdot 5 \times 12) \times \cdot 785 = 137\cdot 8 \text{ lb. per square inch,}$$

or, to all intents and purposes, exactly the same as with petrol, since the lower internal energy balances almost exactly the higher volumetric efficiency.

With alcohol, the same calculation will show that the indicated mean pressure should be 158·3 lb. per square inch, but this is based on a volumetric efficiency of 90 per cent, an efficiency which is not, in practice, realized for reasons already given. If the volumetric efficiency be taken as 82·5 per cent, which corresponds with experimental results under similar conditions, the indicated mean pressure becomes $158{\cdot}3 \times \frac{{\cdot}825}{{\cdot}900} = 145$ lb. per square inch, which is in very close agreement with observed results.

Again, for a 20 per cent weak mixture we must reduce the total internal energy accordingly, but, on the other hand, the thermal efficiency is increased owing to the lower flame temperature; also the volumetric efficiency is reduced very slightly owing to the smaller influence of the latent heat of evaporation for such a mixture. The figures then become:

Fuel.	Indicated Mean Pressure, lb. per sq. in.		Observed I.M.E.P. (Correct Mixture), lb. per sq. in.
	Correct Mixture.	20 per cent Weak.	
Petrol	137·1	118·0	136·0
Benzene	137·8	118·2	136·0
Alcohol	145·0	121·0	143·6

If, now, the mixture be further enriched by the addition of 20 per cent excess of fuel, although the heat liberated per standard cubic inch of mixture will be no greater, the total internal energy will be increased very slightly owing to the greater increase in specific volume when excess of fuel is present; again, owing to the greater latent heat available the volumetric efficiency will be increased slightly, while, finally, the thermal efficiency, based on that portion of the fuel which is burnt completely, will be slightly higher on account of the lower flame temperature; the net result of all these changes will be that the indicated mean pressure will, in all cases, except that of benzene, be some 4·5 per cent greater than with a correct mixture. In the case of benzene, however, owing to the small change in specific volume, the gain in power by using a

20 per cent rich mixture will be only about 3 per cent. If the mixture be enriched beyond 20 per cent excess of fuel, there will be no further gain in power on petrol or benzene owing to the slower rate of burning, but, on alcohol, there will still be a small further gain on account of its high latent heat and large increase of specific volume.

The indicated horse-power obtainable from the cylinder under consideration will be as follows :—

Fuel.	I.H.P.		
	20 per cent Weak.	Correct.	20 per cent Rich.
Petrol	15·75	18·35	19·15
Benzene	15·8	18·4	19·0
Alcohol...	16·55	19·35	20·2

CHAPTER III

DISTRIBUTION OF HEAT IN A HIGH-SPEED FOUR-CYCLE ENGINE

It is usual to express the distribution of heat in an Internal-Combustion engine in terms of the proportion converted into indicated horse-power, the proportion yielded up to the cylinder walls, and, lastly, that rejected to the exhaust; the latter item being the residue after deducting from the total heat of the fuel the two former categories; it generally includes also the losses by radiation. This form of subdivision is perfectly legitimate so long as it is clearly understood that it is no more than a convenient form in which both to measure and to express the heat distribution; indeed, it is practically the only form in which it can be directly measured. Many engineers, however, still appear to be under the impression that it represents the true disposal of the available heat of the fuel, and, acting on this supposition, are often sadly misled.

The proportion of the total heat of the fuel converted to useful horse-power can be determined readily enough and quite accurately from the known heat supplied and the known horse-power developed from it.

The heat yielded up to the cylinder walls and carried away by the cooling water can also be determined fairly accurately; it must be understood, however, that it includes:

(1) The heat given up by radiation, conduction, convection, &c., during the period of combustion.

(2) The heat given up during the expansion period.

(3) The heat given up during the exhaust stroke.

It is necessary to examine each of these sources separately.

(1) **Heat lost during Combustion.**—The period of combustion as distinct from expansion is relatively a short one, but it is one during which the ruling temperature in the combustion chamber is very high indeed, i.e. between 4200° F. and 4500° F. in the case of most volatile liquid fuels such as petrol, benzol, &c. Also it is

a period during which the gases within the combustion chamber are in a state of violent commotion so that heat is conveyed very readily by convection, &c.

Now, if by any means the loss of heat to the cylinder walls during this period could be suppressed, such heat could be converted into indicated horse-power at an efficiency corresponding to the efficiency of expansion alone (i.e. exclusive of the negative work done during compression), which in an engine with a compression ratio of 5 : 1 is roughly about 40 per cent. The remaining 60 per cent of the heat so recovered would, in any event, be rejected to the exhaust after expansion.

(2) **Heat lost during Expansion.**—Loss of heat during the expansion stroke may or may not be serious, depending upon the stage in the expansion stroke at which it is lost. Loss of heat at the very commencement of the expansion stroke is almost equally as serious as that lost during the combustion period, because had its loss been suppressed it would have been utilized at an efficiency corresponding to nearly the full ratio of expansion, whereas heat lost during the latter part of the expansion stroke is of very little moment, for even had its loss been suppressed it could have done but little useful work during the remainder of the expansion stroke and nearly the whole of it would have been rejected to the exhaust in any case.

At first sight it would appear that, owing to the higher temperatures and pressures ruling at the beginning of the expansion stroke, the loss of heat will be much greater during the earlier period, but against this it must be remembered that, as the expansion proceeds and the piston descends, an increasing area of cold cylinder barrel is exposed. Also, owing to dissociation and subsequent recombination, the fall in temperature during the expansion stroke is nothing like so great as it might appear ; the final temperature, even with a compression ratio of 5 : 1, being still well over 3000° F.

From the above considerations it will be seen that, though it is customary to yoke together the heat lost during combustion and expansion as though its influence during each period were the same, it is most certainly inaccurate and misleading to do so. Of the average heat loss during expansion, probably only about 20 per cent could be converted into useful work and the remaining 80 per cent would be rejected to the exhaust.

(3) **Heat lost during the Exhaust Stroke.**—Although during the exhaust stroke the temperature of the gases is very much lower, yet heat is given up to the cooling water with great rapidity during

this period, for in addition to the normal heat flow to the cylinder walls the hot gases are issuing at an exceedingly high velocity past the exhaust valve and through a short length of exhaust pipe which is always included in the cylinder jacket and cooled by the circulating water, consequently of the total heat carried away by the cooling water at least one-half and often more than half is given up during the exhaust period. Now the whole of the heat taken up during the exhaust stroke, by far the bulk of that taken up during expansion, and about 60 per cent of that taken up during combustion, should have been debited to the exhaust loss account.

In addition, of the total heat carried away by the cooling water, a very substantial proportion is generated by the friction of the piston on the cylinder walls.

It is interesting to take a specific example and to trace out as accurately as it is possible so to do the true gain in efficiency which would be effected if all heat loss to the cylinder walls were completely suppressed. Let us take, as a fair average example, the case of a well-designed and efficient engine with a compression ratio of 5 : 1 in which

32 per cent of the total heat of the fuel is converted into useful work on the piston,
28 per cent of the total heat of the fuel is carried away by the cooling water,
40 per cent of the total heat of the fuel remains and is accounted as lost to exhaust radiation, &c.

Of the total heat carried away by the cooling water, approximately 6 per cent will be lost to the walls of the cylinder during the combustion period, about 7 per cent will be yielded up during expansion, and the remaining 15 per cent during the exhaust stroke. Of the 6 per cent lost during the combustion period roughly about 40 per cent would appear as useful work or 2·4 per cent of the total heat of the fuel. Of the 7 per cent lost during expansion, somewhere about 20 per cent would be utilized or 1·4 per cent of the total heat of the fuel. Of the 15 per cent lost during the exhaust stroke, no part could have been utilized. We find therefore that, although 28 per cent of the total heat of the fuel has been carried away by the cooling water, only 3·8 per cent could have been converted directly into useful work on the piston, and the efficiency of the engine would be increased from 32 per cent to 35·8 per cent only, a gain of barely 12 per cent. Nor is this all, for, had all loss of heat to the cylinder walls been suppressed, the temperature of the working fluid would necessarily have been correspondingly higher, with the result that the losses due to the increase both of specific heat and of *dissociation at the higher temperatures would be increased substantially, and the net gain would be very small, probably only from 32 per cent to about 34·5 per cent or possibly 35 per cent.*

These figures show clearly how relatively small a part the loss of heat to the cylinder walls plays in an Internal-Combustion engine, and how misleading it may be to assess that loss of heat by the heat carried away by the cooling water. As a first approximation it is fairly correct to assume that, of all the heat carried away by the cooling water of a cylinder, little more than 10 per cent could actually be converted directly into useful work.

Tables I., II., and III. show the distribution of heat as found in the author's $4\frac{1}{2}$ in. × 8 in. variable-compression research engine shown in figs. 3 and 4, under several different conditions. These figures were all obtained under circumstances which ensure a very high degree of accuracy.

Group A.—Distribution of heat at different speeds with wide open throttle. In all cases the mixture strength was approximately 10 per cent weak, the jacket temperature was kept constant at 140° F., and the heat input to the carburettor was maintained at 0·0433 B.Th.U. per revolution.

The three sets of tests in this group were run, one on ethyl alcohol 95 per cent, and the other on petrol, sample (A), at a compression ratio of 3·8 : 1, while the third set was run on ethyl alcohol 95 per cent at a compression ratio of 7 : 1.

It will be observed :

(1) That owing to the lower mean cycle temperature the thermal efficiency obtained with alcohol is substantially higher than that obtained with petrol at the same compression ratio.

(2) For the same reason, the proportion of heat carried away by the cooling water is less.

(3) That the thermal efficiency is affected very little by a wide variation in speed.

(4) That the proportion of the heat taken away by the cooling water falls slightly as the speed is increased.

The results shown in Group B were observed under the following conditions. The engine was run at a constant speed of 1500 R.P.M., corresponding to a piston speed of 2000 ft. per min., and the load was varied by throttling, the mixture strength being kept constant throughout at about 10 per cent weak, while the heat input to the carburettor was maintained at 0·0433 B.Th.U. at full load and

TABLE I- GROUP A

Fuel, Ethyl Alcohol 95 per cent

Compression ratio 3·8 : 1

R.P.M.	975	1300	1500	1700
Piston speed ft. per min. ...	1300	1733	2000	2266
Heat to I.H.P., per cent ...	26·9	27·0	26·9	27·0
,, cooling water, per cent	25·1	24·7	24·4	24·2
,, exhaust, radiation, &c.	48·0	48·3	48·7	48·8
Total heat	100·0	100·0	100·0	100 0

Fuel, Petrol, sample (A)

Compression ratio 3·8 : 1

R.P.M.	975	...	1500	1700
Piston speed, ft. per min. ...	1300	...	2000	2266
Heat to I.H.P., per cent ...	25·9	...	26·1	26·1
,, cooling water, per cent	30·4	...	28·0	27·0
,, exhaust, radiation, &c.	43·7	...	45·9	46·9
Total heat	100·0	...	100·0	100·0

Fuel, Ethyl Alcohol 95 per cent

Compression ratio 7 : 1

R.P.M.	975	1300	1500	...
Piston speed, ft. per min. ...	1300	1733	2000	...
Heat to I.H.P., per cent ...	37·6	38·1	38·3	...
,, cooling water, per cent	25·4	24·3	23·9	...
,, exhaust, radiation, &c., per cent	37·0	37·6	37·8	...
Total heat	100·0	100·0	100·0	...

reduced proportionately as the load was reduced. The circulating water temperature was maintained constant throughout at 140° F.

Table II—Group B

Fuel, Ethyl Alcohol 95 per cent
Compression ratio 5·45 : 1
R.P.M. 1500

Percentage of maximum I.H.P.	100	80	60	40
Heat to I.H.P., per cent ...	34·8	35·0	35·0	34·8
„ cooling water, per cent	24·1	26·0	29·2	33·0
„ exhaust, radiation, &c., per cent	41·1	39·0	35·8	32·2
Total heat	100·0	100·0	100·0	100·0

Fuel, Petrol, sample (A)
Compression ratio 5·45 : 1
R.P.M. 1500

Percentage of maximum I.H.P.	100	80	60	40
Heat to I.H.P., per cent ...	33·5	34·0	34·1	33·5
„ cooling water, per cent	26·5	28·2	31·8	35·5
„ exhaust, radiation, &c., per cent	40·0	37·8	34·1	31·0
Total heat	100·0	100·0	100·0	100·0

From these tests it will be observed that:

(1) Provided that, as in the above cases, the time of ignition is advanced as the weight of charge is reduced, and therefore the relative proportion of residual exhaust products is increased, the thermal efficiency remains practically constant over a wide range of load.

(2) That as the load is reduced a greater proportion of the exhaust heat appears in the cooling water.

Lastly, a set of tests were run on hydrogen (Table III.). On this fuel alone it is possible to reduce the power output over a wide range y controlling the supply of fuel only, i.e. by qualitative governing, ithout relying upon stratification. These tests are particularly nteresting as they are, it is believed, the only experiments which ave yet been carried out in which the power output has been varied olely by varying the mean mixture strength of the working fluid.

Table III

Fuel, Hydrogen Gas
Compression ratio 5·45 : 1
R.P.M. 1500
Mixture strength at maximum load 10 per cent weak

Percentage of maximum I.H.P.	100	80	60	40
Heat to I.H.P., per cent ...	33·3	35·6	38·2	40·0
„ cooling water, per cent	23·6	24·9	25·3	28·6
„ exhaust, radiation, &c., per cent	43·1	39·5	36·5	31·4
Total heat	100·0	100·0	100·0	100·0

In these tests it will be observed that:

(1) The thermal efficiency increases rapidly as the load is reduced, due to the lower mean temperature.

(2) The proportionate heat supply to the cooling water increases slightly as the load is reduced, but at nothing like the rate shown in the previous tables, when governed quantitatively.

It is interesting to compare the results obtained on hydrogen with those found in the previous group. In the latter case the load is varied by varying the weight of working fluid per cycle, the temperature remaining practically constant, while in the tests with hydrogen the weight remained constant and the temperature was varied. As might be expected when the temperature is varied, so the efficiency follows suit, rising as the mean cycle temperature falls. Incidentally, also, the results obtained in Group B serve to illustrate how small a part direct heat loss to the cylinder walls actually plays, for as the weight of charge is reduced at nearly constant temperature, the *relative* heat loss must be increased substantially and yet the thermal efficiency remains unchanged. That quite a substantial change in heat loss has so little effect on the thermal efficiency illustrates how small, in itself, this whole source of loss must be—

that it actually effects no reduction at all in the thermal efficiency is probably due to the fact that as the weight of charge is reduced, so is the proportion of residual exhaust products increased, with the result that the temperature is slightly lower, and this very slight lowering of the flame temperature results in a gain in efficiency sufficient just to balance the increased relative heat loss.

The conclusions to be drawn from the preceding observations and tests are:

(1) That the direct loss of heat to the cylinder walls plays a comparatively insignificant part in the performance of an Internal-Combustion engine, and that even were the whole of this loss completely suppressed, the gain in power output and efficiency would be equivalent only to the conversion of an extra 2·5 to 3 per cent of the heat of the fuel into useful work.

(2) That, of the total heat carried away by the cooling water, only a very small proportion could be converted into useful work, and by far the bulk would appear in the exhaust.

(3) That when running with wide open throttle, the heat flow to the cooling water is very nearly proportional to the speed of the engine, i.e. the proportion of the total heat input remains nearly constant.

(4) When the load is reduced by throttling a much larger proportion of the heat rejected to the exhaust is taken up by the cooling water, before the gases enter the exhaust pipe.

It is frequently stated that the relative inefficiency of a combustion chamber with a large surface volume ratio is due to the increased heat losses. From the above it is clear that this cannot have any great influence. The most probable cause of the comparative inefficiency is, as shown in Chapter IV, that, owing to the lack of turbulence in such types of combustion chamber, an appreciable portion of the charge clings to and is so chilled by the walls that it either escapes combustion altogether or burns so late in the expansion stroke as to be of little value.

It was mentioned earlier that the division of the heat supplied into three parts is but a convenient way of expressing simply a distribution which has hitherto defied detailed accurate analysis. It will be appreciated that the indicated horse-power is readily obtained to a fair degree of accuracy by motoring the engine immediately after measurement of the brake performance. This is a process now familiar to all. It is found in practice that the amount by which the piston friction when motoring the engine falls short of the friction under gas pressure and heat stress is approximately equal to the increase in pumping loss under motoring conditions, due to the extra work required to expel the exhaust air. The addition of the motoring horse-power to the brake horse-power, therefore, gives a very close estimate of the indicated useful work on the piston, and the resulting value is usually adopted as a means of calculating the Indicated Thermal Efficiency of the engine.

It must not be forgotten, however, that this value includes, quite properly, the work due to piston friction, which is dissipated as heat and carried away, partly in the cooling water, partly by radiation. Measurement of the heat taken up by the cooling water therefore includes a small quantity of heat which has already been accounted for in the indicated horse-power.

For this reason it is becoming customary to divide the heat supplied into three somewhat different quantities as follows: the brake horse-power; the heat loss to the cooling water and in radiation; and the heat loss to exhaust.

The heat lost in radiation may be measured by motoring the engine at the running speed and temperatures, by means of a swinging field dynamometer with the exhaust and inlet pipes blanked off. The temperature of the water and lubricating oil must be maintained by immersion heaters and the required heat input measured. The total loss of heat by radiation is then obtained by adding the power required to motor the engine converted to heat units to the measured input to the heaters. This method gives a more accurate determination than the static method, because, when running, the flywheel windage increases the heat loss from the engine crankcase, and this effect may be appreciable. Against this must be set a small amount of power registered by the dynamometer in respect of flywheel windage. This may be allowed for, if desired, but its value is quite small in relation to the magnitude of the measured losses, which may amount to as much as 20 or 25 per cent of the total losses in small engines.

The Table on p. 96 gives the heat distribution divided in this way for a number of typical engines, together with the conditions of operation.

The curves of fig. 37 are summary curves giving gross heat loss (cooling water and lubricating oil plus radiation losses) plotted against specific quantity of fuel supplied per litre cylinder capacity per hour. The curves represent the average of a number of engines unsupercharged and supercharged. For poppet-valve petrol engines

	Percentage of Heat Input.		
	Heat to B.H.P.	*Heat to Cooling Water and Radiation.*	*Heat to Exhaust.*
Normally aspirated petrol engine, C.R. 5·5, 5 lb. fuel/litre/hour.	30	35	35
Heavily Supercharged Petrol Engine, C.R. 4·3, 25 lb. fuel/litre/hour.	22	18	60
Normally aspirated Diesel Engine, C.R. 16, 5 lb. fuel/litre/hour.	32·5	33	34·5
Heavily Supercharged Diesel engine, C.R. 9, 25 lb. fuel/litre/hour.	27	19	54

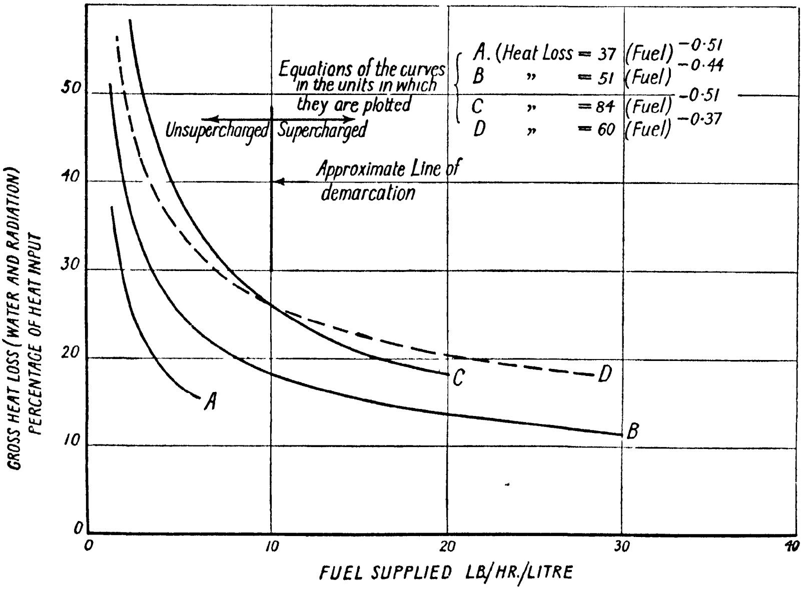


Fig. 37.—Mean Curves of Heat Loss to Water, Lubricating Oil, and Radiation

A. Diesel Engines:		Open type chamber—practically no air swirl.
B. Petrol	,,	Sleeve valve engines—moderate turbulence.
C. ,,	,,	Turbulent cylinder heads.
D. Diesel	,,	Using intense air swirl.

cold jackets will only be 4 per cent of 10 per cent, or about 0·4 per cent. With a very badly designed combustion chamber it might conceivably amount to as much as 1 per cent, but in any case it is but a trifling amount.

With regard to the variation in volumetric efficiency the difference here is much more marked and is in the reverse direction. With hot cylinder walls at 300° F., the rise in temperature of the working fluid due to contact with the inlet valve and cylinder walls during its entry will amount to about 80° F. With cold water it will be about 55° F., assuming that the mean surface temperature of the walls is, in each case, about 90° F. higher than that of the jacket water. Experiments quoted elsewhere show that with a cylinder of normal design the change in the temperature rise of the entering gases is about one-sixth of that of the cylinder walls. The mean absolute temperature of the working fluid after its entry to the cylinder may be taken as about 700° F. absolute. The weight of charge taken into the cylinder and therefore the power output will be proportional to the absolute temperature, so that if this is reduced by 25° F. the weight of charge taken into the cylinder, per cycle, will be increased in the proportion of $\frac{700}{675}$, or about 3·75 per cent. From the above considerations it will be seen that while the power and efficiency may be reduced by from 0·4 per cent to 1 per cent, due to increased heat losses, the weight of charge taken in will be increased by about 3·75 per cent, the net result being a gain of about 2·75 to 3·25 per cent in the indicated horse-power by reducing the temperature of the cooling water from boiling to about 72° F. or what is usually termed stone cold.

So far as the indicated horse-power is concerned, therefore, any reduction in the temperature of the cooling water involves an appreciable increase in the power developed in the cylinder. Fig. 42 shows the mean results of a large number of actual tests, from which it will be seen that the observed results agree very closely with the above deductions.

The third factor, namely piston friction, plays a very important part. In Chapter IX, dealing with piston design, it will be emphasized that piston friction is dependent in a very large measure on the viscosity of the lubricating oil, and therefore upon the temperature of the cylinder walls—the colder the walls, the higher the viscosity of the oil clinging to them, and therefore the greater the friction. In the case of pistons with large bearing surfaces and heavy inertia pressures the difference in friction between hot and cold jackets may amount to as much as 8 per cent of the indicated horse-power of the engine, in which case the gain in indicated horse-power from the use of cold circulating water would be completely swamped by the extra friction, and the net result would be a loss of about 5 per cent in brake horse-power. On the other hand, with a very light piston, with the minimum of bearing surface, the difference may be as little as 3 per cent, in which event the brake horse-power of the engine will be independent of the jacket water temperature. Quite recently tests have been carried out on three single-cylinder engines, each of about the same size, in which the carburettor and induction system were heated independently of the circulating water. One of these engines was fitted with a cast-iron trunk piston of orthodox design, one with a very light aluminium slipper-type piston, and one with a cross-head piston, in which the friction of the piston on the cylinder walls is practically negligible. The results obtained are given in the table on p. 105.

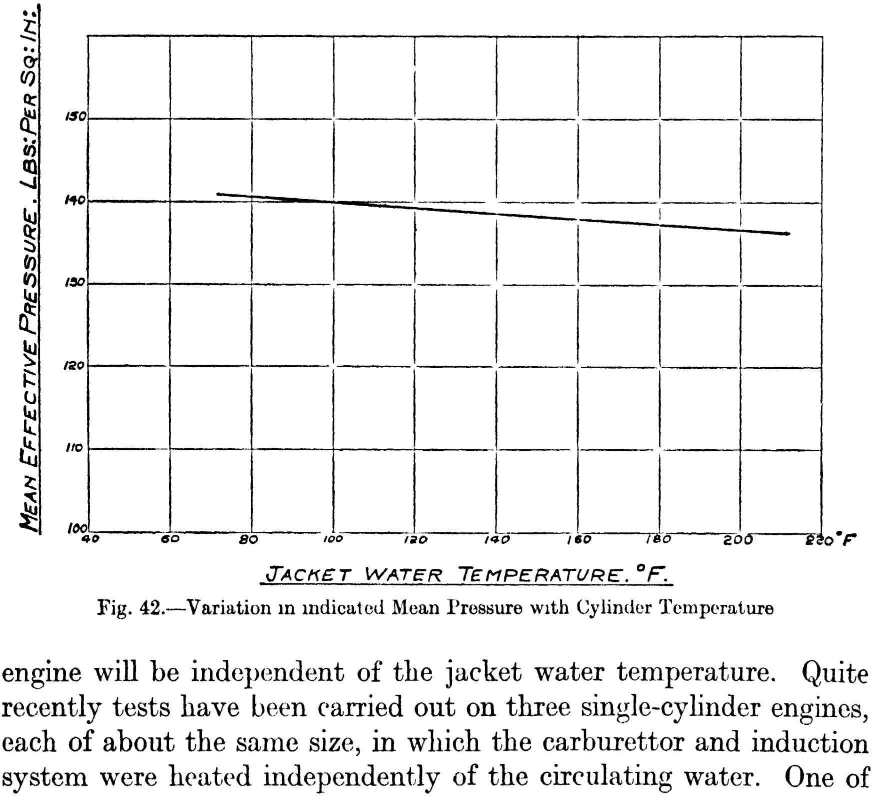


Fig. 42.—Variation in indicated Mean Pressure with Cylinder Temperature

In the case of the engines used for tanks which were fitted with cross-head pistons and independently heated induction systems, it was invariably found that the brake horse-power was from 1·5 per cent to 2 per cent higher with "cold" jacket water, i.e. about 80° F., than with "hot", i.e. about 180°–200° F. The difference in power was quite appreciable in service, and tank drivers frequently expressed surprise that their engines appeared to be more "lively" when the cylinders were cold.

To sum up, apart from any question of carburettor temperature, the power output of an engine may increase or decrease as the jacket temperature is raised, depending upon the piston friction; if the piston friction is high, the power output will increase; if low, it will not increase. The increase in heat loss over the most extreme conditions is so small as to be negligible. The increase in volumetric efficiency is comparatively large, but is usually not sufficient to balance the increase in piston friction, hence the loss of power so often observed.

	B.H.P. at 1200 R.P.M.			
	Engine No.	Jacket Water, 212° F.	Jacket Water, 80° F.	Difference per cent
(1) Cast-iron trunk piston ..	B. 8	27	25·7	−5
(2) Aluminium slipper-type piston	E. 35	28·3	28·3	Nil
(3) Cross-head piston	T.S. 1	29·4	30·0	+2

In most cases, however, the temperature of the carburettor and induction system is dependent also upon the jacket-water temperature, and when operating with fuels of low volatility the variation of temperature of these parts may play a supremely important part, particularly if the distribution is inherently defective or the mixture is rather weak.

Gas Velocity and Indicated Mean Pressure.—From analysis of a very large number of observations and test results, the author has found that the best all-round compromise between such various conflicting conditions as (1) the attainment of the highest possible volumetric efficiency, (2) the attainment of the necessary degree of turbulence, (3) the reduction of the fluid pumping losses to the lowest possible limit, is achieved when the mean gas velocity through the valves is in the neighbourhood of 130 ft. per second. Clearly from the point of view of conditions (1) and (3) it is desirable to keep the velocity as low as possible, while to fulfil condition (2) a high gas velocity is required. The gas velocity, as is customary, is expressed in terms of feet per second through the valve opening *on the assumption (a) that the valve is wide open throughout the entire stroke; (b) that the mean and not the maximum piston speed is taken into account.*

There is now abundant evidence that where the valves open directly into the main body of the combustion chamber a somewhat higher gas velocity may be employed without impairing the volumetric efficiency; on the other hand, with such a combustion chamber free from pockets or recesses turbulence will be better maintained and a lower gas velocity will suffice. In other words, when the valves are so placed that they open freely into the combustion space, both the efficiency and the mean pressure are less dependent upon gas velocity. In practice, however, the size of the valves and valve passages is largely controlled by mechanical limitations. Practical experience indicates that, when valves are fitted directly in the cylinder heads, only as much area of opening should be given as can be provided without allowing the symmetry and compactness of the combustion chamber to suffer. It would seem preferable, therefore, to increase the gas velocity up to as high as 160 ft. per second, rather than to distort the combustion chamber in order to accommodate larger valves.

The valve-opening diagram is clearly a very important factor, and in the absence of precise information it must be assumed that the valve-opening diagram is in each case that best suited to the general design of the engine and its pipe-work.

The author has now a great deal more information on this subject, derived from a very large number of engines tested in his own laboratory and under his own observation. Further, since all these engines were designed on the same basis having the same valve-opening diagrams, and all running at sufficiently high speeds to eliminate any appreciable variation in the heat losses, the results are directly comparable; and the deductions to be drawn from them are probably quite reliable. The results of some of these tests are shown in fig. 43, from which it will be seen that, while both the mean pressure and fuel consumption differ very considerably, depending both upon the compression ratio and the efficiency of the combustion chamber, the relation between mean pressure, efficiency, and gas velocity is substantially the same in all cases.

Again, a series of experiments was carried out on the special single-cylinder research engine illustrated in figs. 3 and 4; this engine is fitted with two inlet and three exhaust valves. Provision is made for throwing any of the valves out of operation.

The results obtained with various combinations of valves in action are shown in fig. 44 (p. 108).

Many useful deductions can be drawn from the results of this last series, the more so because: (1) The tests were carried out under conditions ensuring an exceptional degree of accuracy; (2) the area of valve opening, and therefore the gas velocity, could be altered without disturbing any other condition.

All experience serves to indicate that the velocity through the exhaust valves may be as much as 50 per cent higher than through the inlet valves without causing any appreciable resistance or other loss. Although the same weight of working fluid has to pass through both the inlet and exhaust valves in much the same time the conditions under which it passes are very different. In the first place, at the time when the exhaust valves open, the pressure in the cylinder is relatively high, generally from 50 to 70 lb. per square inch above atmosphere. The sudden release of the gases at this high pressure results in the setting up in the exhaust pipe of a very high velocity, and the kinetic energy acquired by the gases in this pipe assists greatly in the withdrawal of the remainder from the cylinder. Unless, therefore, the silencer offers any undue resistance the energy

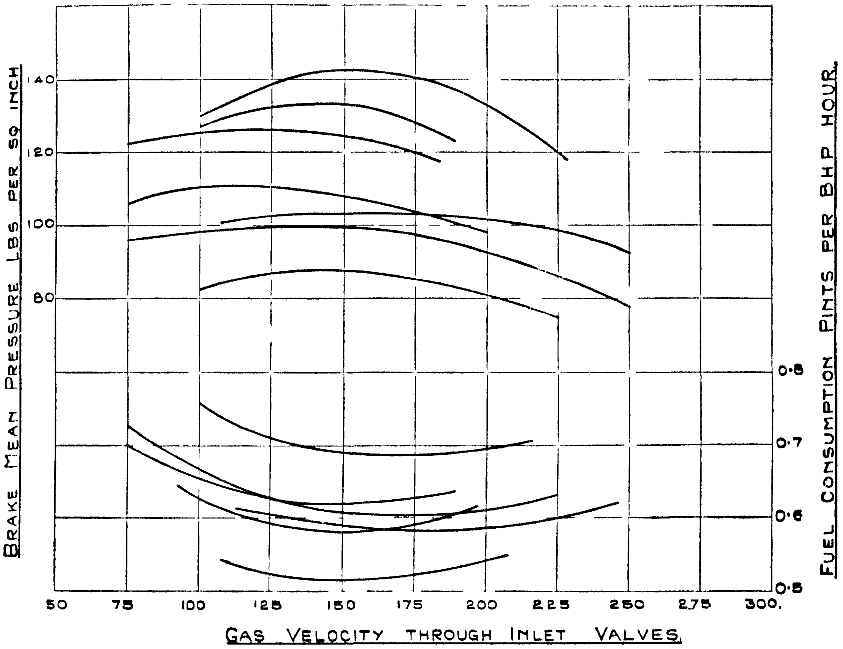


Fig. 43.—A Selection of Curves showing M.E.P. and Fuel Consumption on a Gas-Velocity Basis

TESTS WITH ONE OR MORE VALVES THROWN OUT OF OPERATION

I.H.P. (Compression ratio 6 : 1 in all cases.)

Number of Valves in Operation and Mean Gas Velocity.

R.P.M.	Column No. 1.			Column No. 2.			Column No. 3.			Column No. 4.		
	No. of Valves 2 inl. 3 exh.			No. of Valves 2 inl. 2 exh.			No. of valves 2 inl. 1 exh.			No. of valves 1 inl. 3 exh.		
	Gas velocity, ft. per sec.		I.H.P.	Gas velocity, ft. per sec.		I H.P.	Gas velocity, ft. per sec.		I.H.P.	Gas velocity, ft. per sec.		I H.P.
	Inlet.	Exhaust		Inlet.	Exhaust.		Inlet.	Exhaust.		Inlet.	Exhaust.	
900	66·1	86·6	19·9	66·1	137·2	20·0	66·1	235·0	18·8	132·2	86·6	19·1
1100	81·0	106·0	24·5	81·0	168·0	24·55	81·0	287·5	22·9	162·0	106·0	23·0
1300	95·5	125·0	29·4	95·5	198.0	29·4	95·5	339·0	26·0	191·0	125·0	27·0
1500	111·0	144·5	34·5	111·0	228·5	34·4	111·0	391·0	26·7	222·0	144·5	30·7
1700	124·8	163·0	39·8	124·8	259·0	39·3	124·8	443·0	26·4	249·6	163·0	33·8
1900	139·6	183·0	45·1	139·6	290·0	44·3	139·6	496·0	25·2	279·2	183·0	36·0

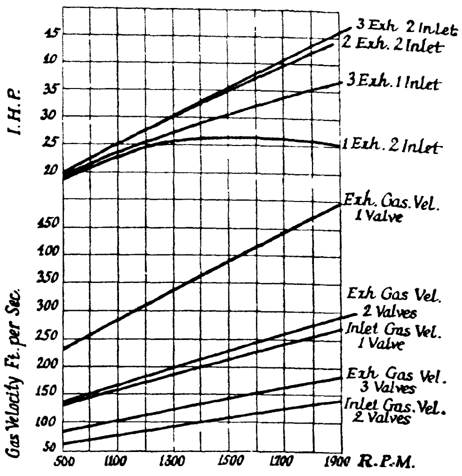


Fig. 44.

expended in driving out the exhaust products is almost negligible, and, within limits, is almost independent of the gas velocity.

While distinct evidence of wire-drawing accompanied by a rapid increase in the fluid pumping losses becomes apparent so soon as the inlet gas velocity exceeds about 150 to 160 ft. per second, there is no evidence of any measurable fluid resistance, at all events until the exhaust gas velocity exceeds about 240 ft. per second.

One reason for this is that owing to their high release pressure the bulk of the exhaust products is discharged while the piston is more or less at rest. It must also be remembered that while a back pressure of, say, 1 lb./in. on the exhaust stroke only decreases the effective mean pressure by that amount, a negative pressure of 1 lb./in. at the end of the suction stroke will reduce the mean pressure by $\frac{1}{14\cdot7}$ or nearly 7 per cent, a much greater loss.

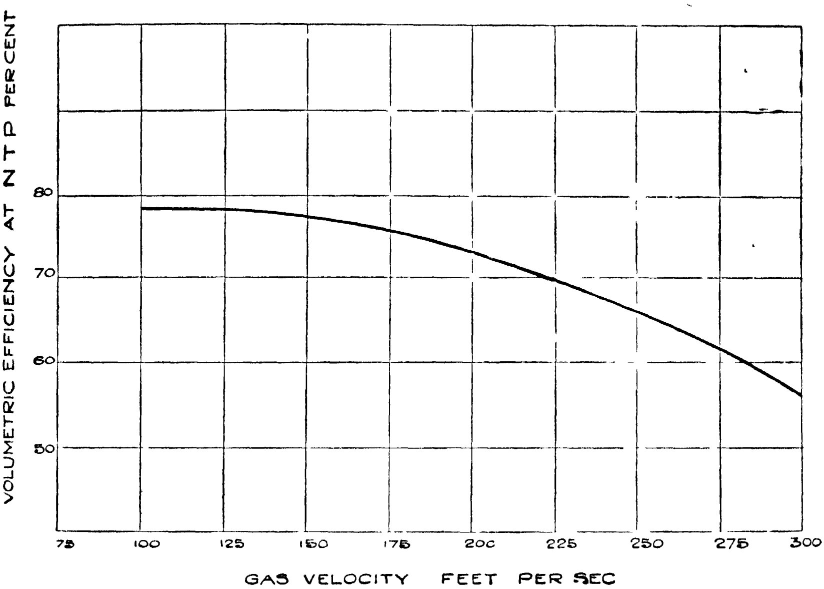


Fig. 45.—Volumetric Efficiency and Gas Velocity

From the more reliable data now available it is possible to construct a curve giving the relation between inlet gas velocity and volumetric efficiency with a very fair degree of accuracy. As already explained, however, the volumetric efficiency is also influenced to a very large extent by the vaporization of fuel within the cylinder and upon the degree of pre-heating before entry; it is therefore dependent upon these factors also. The curve shown in fig. 45

gives to a close approximation the relation between the volumetric efficiency obtained and inlet gas velocity for an efficient and up-to-date design of petrol engine with normal valve timing; for an engine running on benzol it is about 2·5 per cent too low, and for alcohol about 8 per cent too low, provided, of course, that the amount of pre-heating is the same for each fuel. The curve also assumes that the induction piping and carburettor offers no abnormal resistance, and that not more than four cylinders draw their supply from any one carburettor.

To sum up, it is clear that all the evidence available to-day tends to confirm the assumption that the most satisfactory compromise is attained when the gas velocity through the inlet valves is from 130 to 160 ft. per second, depending upon their position in the combustion chamber; the gas velocity through the exhaust valves may, however, be as much as 50 per cent greater without any ill effect.

Influence of Cylinder Size on Power and Efficiency.—For equal compression ratios the indicated power and efficiency of an engine is influenced by two factors: (1) Incomplete combustion of the layer of working fluid which clings to the walls of the combustion chamber; (2) direct loss of heat to the walls of the combustion chamber.

Both depend clearly upon the relation between the surface of the combustion chamber and its volume, but the former depends also upon the thickness of the layer adhering to the walls, which in turn is dependent upon the degree of turbulence or scouring effect. Both these sources of loss are reduced to a minimum when the combustion chamber is as compact and symmetrical as possible, for not only is the surface/volume ratio then reduced to a minimum, but the scouring effect of turbulence has the fullest possible play.

Assuming that the combustion chamber is similar in all cases, then it is evident that the relation of surface to volume will be reduced as the size of the cylinder is increased; since, so long as the proportions are similar, one factor varies as the square of the size and the other as the cube. It follows, therefore, that under similar conditions the power output per unit volume and efficiency of an engine will increase with increase of size. Exactly how much it will increase must depend upon the proportion which these sources of loss bear to the whole; that is to say, the gain in power and efficiency with larger cylinders must depend largely upon the design of the combustion chamber. When the combustion chamber in all cases is as compact and symmetrical as possible, the gain with increase of size will be at a minimum.

It should be emphasized that the above arguments apply for equal compression ratios. When, however, the effect of cylinder size on the incidence of detonation is taken into account, it will be found that the compression ratio of the smaller cylinder can be increased to an extent which will compensate for the greater heat, loss, &c., with the result that at its appropriate compression ratio the performance—viz. mean pressure and efficiency of any petrol engine—is virtually independent of cylinder size.

CHAPTER IV

INFLUENCE OF FORM OF COMBUSTION CHAMBER

Of all the features of design which control both the power output and efficiency of an internal-combustion engine, by far the most important is the form of the combustion chamber. Upon this depends not only the efficiency with which the fuel is burnt, and therefore the power output and efficiency of the engine, but also, to a very large extent, the liability to detonation; and detonation by limiting the compression ratio that may be used sets an additional limit on the efficiency.

The term " Efficiency ", never very closely defined, has, of late years, become much abused. It is quite customary nowadays to describe an engine as " efficient " merely because it is capable, mechanically, of running at a very high speed, though this is clearly a misuse of the term.

While the term " efficiency " is often abused, its true importance is still more often overlooked. It is argued frequently that since fuel is cheap and air still cheaper " efficiency " is of but little practical importance. From the point of view of fuel cost, this is often true, but it cannot too strongly be emphasized that besides determining both the power output and fuel economy the efficiency of an engine is also a measure of its reliability and maintenance of tune, for it must be remembered that the reliability of an engine depends, not so much, it is true, on the proportion of the total heat converted into useful work, but rather upon the proportion of the total heat which is not so converted, and which is left over to make trouble. Fully 90 per cent of the ills from which internal-combustion engines suffer are due, directly or indirectly, to excessive heat flow; directly in the form of cracked cylinders and valve seats, seized or burnt pistons and burnt out valves; in the form of carbonization, or gummed up piston rings, &c., and indirectly in the form of partially carbonized or contaminated lubricating oil, resulting in choked oilways and consequent bearing failures. Now each and all of these sources of trouble, whether they result in complete breakdown or in loss of tune, are attributable to excessive heat flow, and it must constantly be borne in mind that the whole of this destruction is wrought by the flow of waste heat, and that one is really more interested in reducing the amount of heat left behind than anything else. The only possible way in which the waste or residual heat can be reduced is by converting a greater proportion of it into useful work.

Let us consider two engines of the same given cylinder capacity; each, at the same speed, will consume the same weight of air. Whether that air be saturated or supersaturated with fuel, the same amount of heat will be liberated, and we will suppose that it is the equivalent of 100 H.P. in each case. If now one engine has a thermal efficiency of 30 per cent and the other of 20 per cent, it follows that not only will the former engine give 30 H.P. as against 20 H.P. in the other case, but the amount of heat left over to make trouble will be only 70 per cent in the former case and 80 per cent in the latter, so that we can actually get 50 per cent more power from the same engine, and that with a lower heat flow. Now suppose that we run both engines at the same power output, say 20 H.P. To get this power output in the case of the more efficient engine we shall need to liberate only the equivalent of 66·6 H.P. worth of heat, 20 of which will be available as power and 46·6 will be left over. In the case of the less efficient engine we have seen that to get 20 H.P. we have to liberate 100 H.P. worth of heat, and have the equivalent of 80 H.P. left over to make trouble. It will be seen, therefore, that by increasing the efficiency from 20 to 30 per cent, the waste heat is reduced to very little more than one-half, and the more efficient engine will run probably ten times as long at the same power output without trouble or loss of tune. It is on the score, therefore, of reliability and of sustained and consistent performance that the necessity for efficiency is most insistent.

In the design of the combustion chamber the most important considerations are:

(1) The maintenance of the turbulence set up by the gases during their entry;

(2) The position of the ignition plug;

(3) The avoidance of any pockets where the gases may become stagnant;

(4) The provision of a free and unobstructed entry for the gases after passing through the inlet valve.

Turbulence.—The maintenance of turbulence until the time of ignition is probably the most important consideration of all, for upon turbulence depends the rate at which combustion takes place. If the combustible mixture were completely stagnant at the time of ignition, the initial flame brought into life by the passage of the spark across the points of the ignition plug would spread so slowly that, even in a comparatively slow-speed engine, barely half the working fluid would be burnt before the exhaust valve opens. No matter what fuel be used (except possibly hydrogen), we have to rely almost entirely on turbulence or mechanical disturbance to distribute the wan and timid flame rapidly throughout the combustible mixture, and this becomes the more important as the density of the charge is reduced by throttling; for as the density is reduced so is the proportion of residual exhaust products increased, and these, being diluents, tend to lower the flame temperature and so greatly to discourage and retard the process of inflammation. We rely, therefore, entirely upon mechanical disturbance, or turbulence, to speed up the process of combustion and to spread inflammation rapidly throughout the whole bulk of the working fluid.

Apart, however, from the question of accelerating combustion, turbulence plays another and highly important part. In any internal-combustion engine cylinder there is always a layer of working fluid adhering more or less to the cold cylinder walls. This, by its close proximity to cool surfaces, can get rid of its heat so rapidly that it escapes complete combustion. It is upon turbulence that we have to rely, to scour away this layer and to distribute it throughout the combustion chamber. There is little doubt but that the effective thickness of this layer, and its influence upon the power output and efficiency of an engine, depend largely upon the degree of turbulence within the combustion chamber.

Turbulence in itself, so far as the author has been able to ascertain, has no very important influence upon the tendency of the combustible mixture to detonate. If the mixture is completely stagnant and ignited from any one point, the flame will extend from that point very slowly at first, but at a steadily increasing velocity until ultimately it reaches a speed at which the unburnt gas, compressed by the rapidly approaching flame, receives heat from compression, and also by radiation and conduction at a rate in excess of that at which it can get rid of it to the cool cylinder walls, with the result that its temperature is raised above its self-ignition temperature. It then ignites spontaneously and almost instantaneously throughout its whole bulk, thus setting up an explosion or detonation wave. If instead of being stagnant the combustible mixture is in a state of violent disturbance at the time when ignition is started, the whole process is speeded up enormously; but it is the *whole* process that is so speeded up and not any one part of it, with the result that the phenomena are precisely the same, though on a different scale so far as speed is concerned.

It is generally supposed that a combustion chamber having a large surface in relation to its volume owes its low efficiency entirely to direct heat loss to the cylinder walls, but in Chapter III. we have seen that direct heat loss plays a relatively small part in the performance of an engine, and it is far more probable that this form of combustion chamber owes its low efficiency rather to the fact that not only is the area of the layer of gas which clings to the walls and so escapes combustion considerably greater, but, in addition, its thickness also is greater, due to the fact that in such a chamber there is more resistance to the motion of the gases, and the scouring effect of turbulence is therefore less. Although it is not possible to state definitely that this is the case, yet there is a great deal of presumptive evidence in favour of such an assumption; not the least of which being that in any given design of combustion chamber an increase in turbulence, e.g. by increasing the velocity through the inlet valve, will often produce an increase in efficiency without altering in any respect the flow of heat to the cooling water. This, in itself, would not be conclusive evidence, but coupled with a great deal of other indirect evidence it carries a good deal of weight.

Position of the Ignition Plug.—The position of the ignition plug in the combustion chamber is very important, for upon it depends:

(1) The tendency to detonate;

(2) The time taken to complete combustion.

The influence of sparking plug position upon detonation has already been discussed in Chapter II and need not be reiterated here.

Apart from the question of detonation, the distance from the point of ignition to the farthest point in the combustion chamber controls, for an equal degree of turbulence, the time taken for inflammation to spread throughout the working fluid. This is very important, more particularly so on light loads, for, other things being equal, the rate of burning depends upon the flame temperature, and this, in turn, depends upon the proportion of residual exhaust products, *As the load is reduced by throttling, so is the proportion of inert gases increased, and, in consequence, the flame temperature is* reduced and with it the rate of burning. It may be argued that, within limits, the rate of burning matters but little, because it is always possible to advance the time of ignition to suit. This, however, does not quite meet the case, even on full load, for, if the period of combustion be prolonged unduly, the direct heat loss to the cylinder walls at this vital period may begin to assume serious proportions, while when the load varies rapidly, as it does in all forms of motor traction, it becomes almost impossible so to adjust the time of ignition as to suit all conditions.

If the plug is placed centrally in the combustion chamber, a small ignition advance will serve for all conditions of speed or load, and the engine will run at all times at a high efficiency with fixed ignition timing. If, however, the plug is placed in such a position as to leave a long travel for the flame, not only will a considerable ignition advance be required at full load, but the engine will, at all times, be much more sensitive to the time of ignition, and will not run efficiently at light loads unless the ignition is still further advanced. It should be noted that both the time taken to complete combustion and the tendency to detonate depend, for equal degrees of turbulence, upon the absolute, not the relative, distance from the sparking plug to the farthest point in the combustion chamber. Thus, both the time of combustion and the tendency to detonate will be nearly the same in a cylinder of 8-inch diameter with the plug in the centre as in one of 4-inch diameter with the plug at one side, or for similar plug positions both the tendency to detonate and the time of combustion will be greater with the larger cylinder. This factor would militate against the large cylinder were it not for the fact that in the larger cylinder turbulence is the better maintained, the gases are in more rapid motion, and combustion therefore spreads relatively faster. Hence the efficiency does not suffer, though the tendency to detonate necessarily remains greater in the larger cylinder.

In the design of the combustion chamber the three points to be aimed at are:

(1) To provide a chamber into which the valves open directly, so that after entry the gases need turn no corners and so lose their initial velocity;

(2) To arrange that the sparking plug be placed centrally in the combustion chamber;

(3) That there should be no valve or other pockets where the gases may become stagnant.

At first sight it might appear that the ideal combustion chamber would be a true sphere, but although this would be excellent from the point of view both of surface/volume ratio and of turbulence, it would be ideal only if the point of ignition were at the centre, which is obviously impossible. Since the point of ignition must of necessity be situated somewhere on or very near the surface, it follows that the distance which the flame will have to travel will be equal to the full diameter of the sphere, and this will give rise to detonation and necessitate the use of a lower compression ratio, with consequent loss of power and efficiency. Probably the nearest practical approaches to the ideal combustion chamber is that shown in fig. 46.

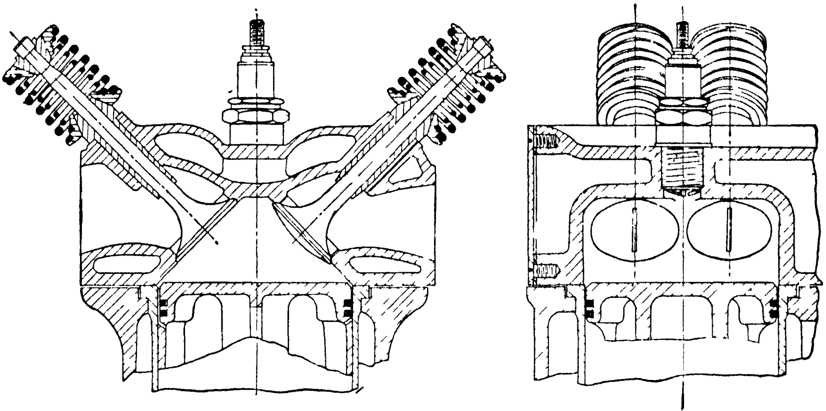


Fig. 46

This form, while almost ideal on thermo-dynamic grounds, is not very convenient from a mechanical point of view, because it necessitates the use of overhead valves operated from two camshafts, for the central position of the sparking plug precludes the use of a central overhead camshaft. It is, however, particularly applicable to sleeve-valve engines, and the fact that it is inherent to this type of engine is, in the author's opinion, one of the strongest arguments in favour of the sleeve valve. For engines required to develop a high power output and efficiency, similar forms of combustion chamber but with the sparking plug placed at the side, are very commonly employed, and are, indeed, practically universal for all aircraft engines. The effect, however, of placing the plug at the side is to increase both the degree of ignition advance and the *tendency to detonate*, and so necessitate the use of a somewhat lower compression ratio and a consequent reduction both in power and efficiency. By using two sparking plugs, however, placed at opposite sides of the combustion chamber, this disadvantage can largely be overcome, for the distance which the flame has to travel from the point of ignition until combustion is completed is then almost halved. In the case of aero-engines where dual ignition is insisted upon, on the score of reliability, this form has become almost a standard one.

The form of combustion shown in fig. 47, in which two horizontal valves are placed in a small cylindrical combustion chamber, is one of the best possible from the point of view both of detonation and combustion efficiency so long as the clearance between the piston and the cylinder head is reduced to the very limit necessitated by mechanical considerations, and so long also as the engine runs at a comparatively low piston speed and therefore does not require large valves. It is therefore a particularly efficient form for short-stroke engines, but though excellent for power output and efficiency it is unfortunately very inconvenient from a mechanical point of view when applied to a vertical engine on account of the valve operation and pipe work.

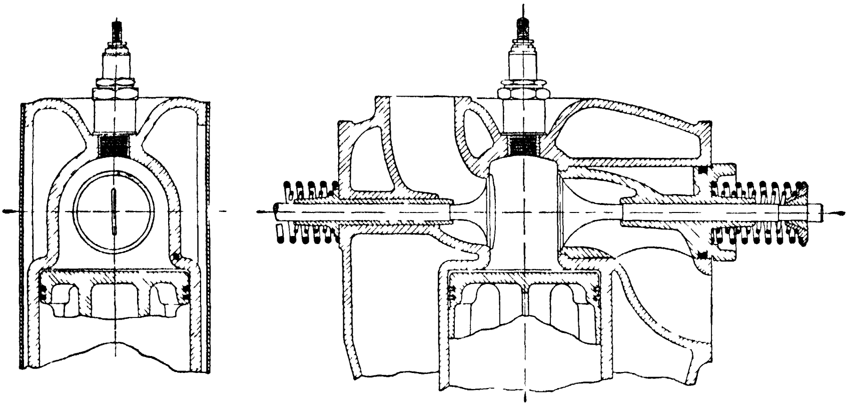


Fig 47

The form shown in fig. 48, in which the inlet valve is placed vertically over the exhaust in a side-pocket, and in which both the piston and cylinder crown are concave, is surprisingly efficient in maintaining turbulence despite the fact that the valves are pocketed, but, as may be expected with so long a travel for the flame, it is very bad from the point of view of detonation and therefore can only be used with a low compression ratio. With this form of combustion chamber and with the sparking plug fitted in the valve pocket, the tendency to detonate is considerable, while with the plug fitted on the side, remote from the valve pocket, it is one of the very worst possible forms, from the point of view of detonation, as indeed might be expected. This form, therefore, necessitates the use of a low compression ratio, but gives a very good power output and efficiency at that compression.

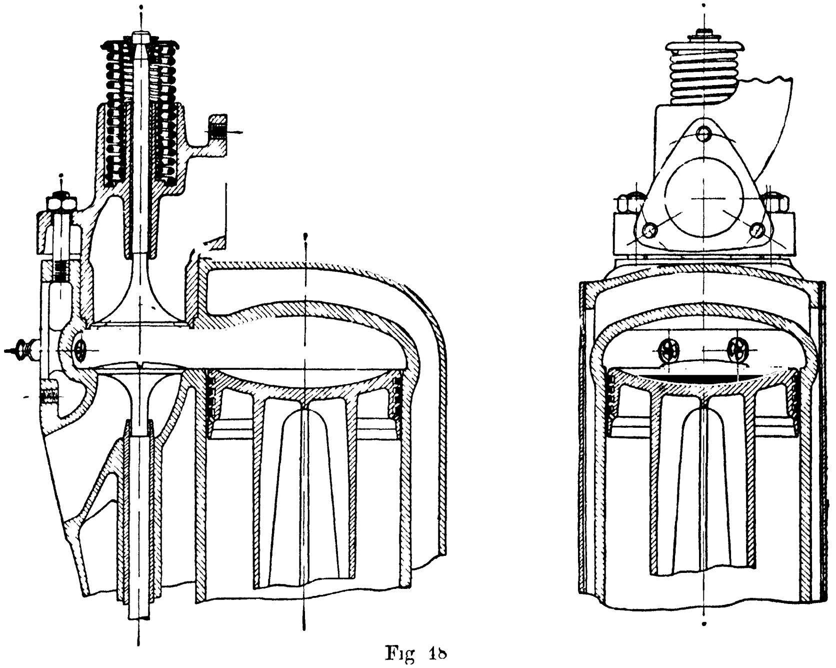


Fig 48

Some ten years ago, and as the result of a good deal of previous experimental work carried out before and during the War, the author patented a form of cylinder head for side-valve engines; an example of which is shown in fig 49; this has come to be known as the "turbulent head", and is now widely used. The primary objects aimed at in this design of head were:

(1) To create additional turbulence during the compression stroke in order to:

(*a*) Increase the rapidity of burning and so obtain both a greater effective expansion ratio and, at the same time, render the engine much less susceptible to ignition timing.

(*b*) Scour away as far as possible the layer of gas which normally clings to the cool surfaces of the combustion chamber and is

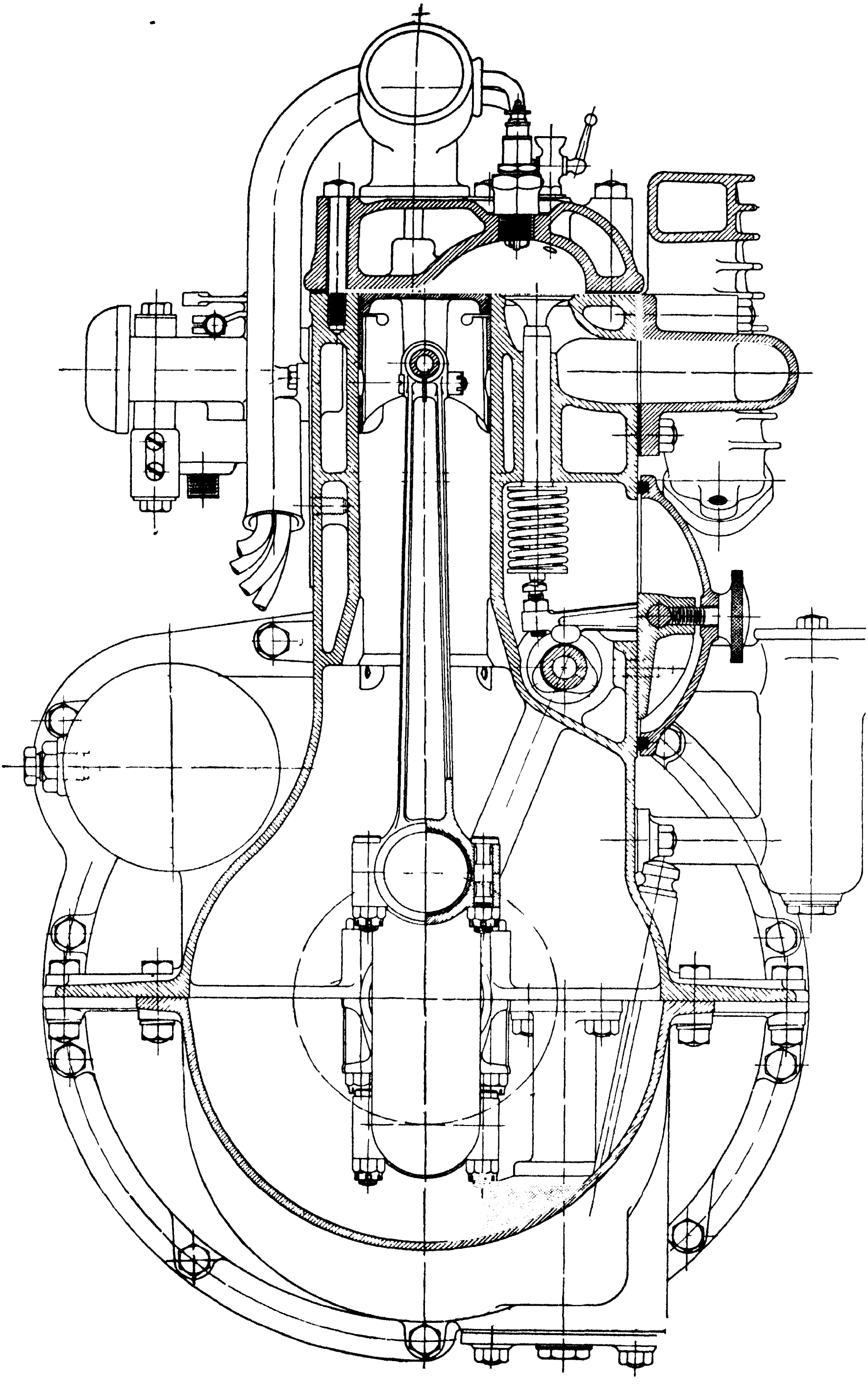


Fig 49

therefore chilled to such an extent as to escape complete combustion either entirely or until so late in the expansion stroke as to be of little value.

(c) Reduce the tendency to detonate by keeping the unburnt gas in rapid motion, thus enabling it the more readily to get rid of the heat of compression by the oncoming flame front and, at the same time, to break up the flame front.

(2) To reduce the length of flame travel from the sparking plug to the farthest point in the combustion chamber by rendering inoperative, so far as detonation is concerned, the part of the combustion chamber over the farthest side of the piston. This was accomplished by bringing the piston into such close contact with the head that the gas between these two relatively cool surfaces was so chilled as to avoid detonation.

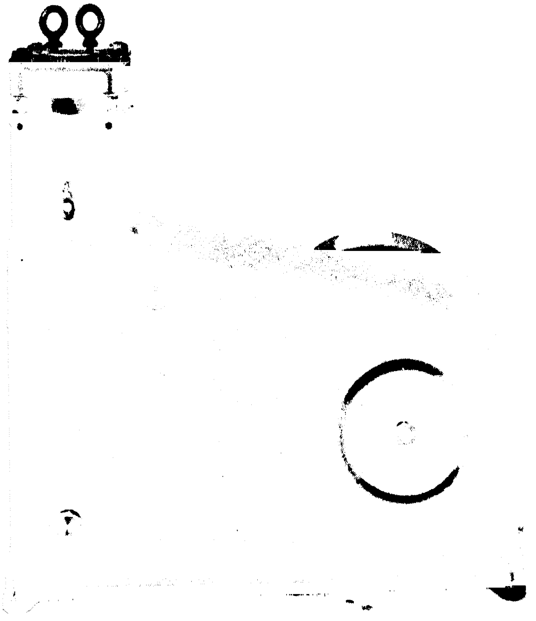


Fig 50

(3) To keep the flame travel as short as possible by placing the sparking plug in a central position.

(4) To reduce, to the minimum, the surface/volume ratio and therefore the heat loss during combustion, though this latter is relatively small.

The above were the intentions of the design at the time it was *first produced; by such means it was found possible to obtain from a side-valve engine a power output and economy at least as good as those previously obtainable from an overhead valve engine of* equal dimensions, with the added advantage of a reduction both in the tendency to detonate and in the " sensitiveness " to ignition timing.

During the past few years the author and his staff have been carrying out a series of investigations in relation both to the effects of turbulence and detonation. For this purpose, the following specially designed apparatus has been employed:

(1) A special single-stroke variable compression engine in which the piston makes one rapid compression stroke and is immediately caught and locked on top dead centre. The charge in the combustion chamber can be fired either by a sparking plug in the ordinary way, or by compression ignition alone. A small fan is fitted inside the combustion chamber to give any desired degree of turbulence. See figs. 50 and 51.

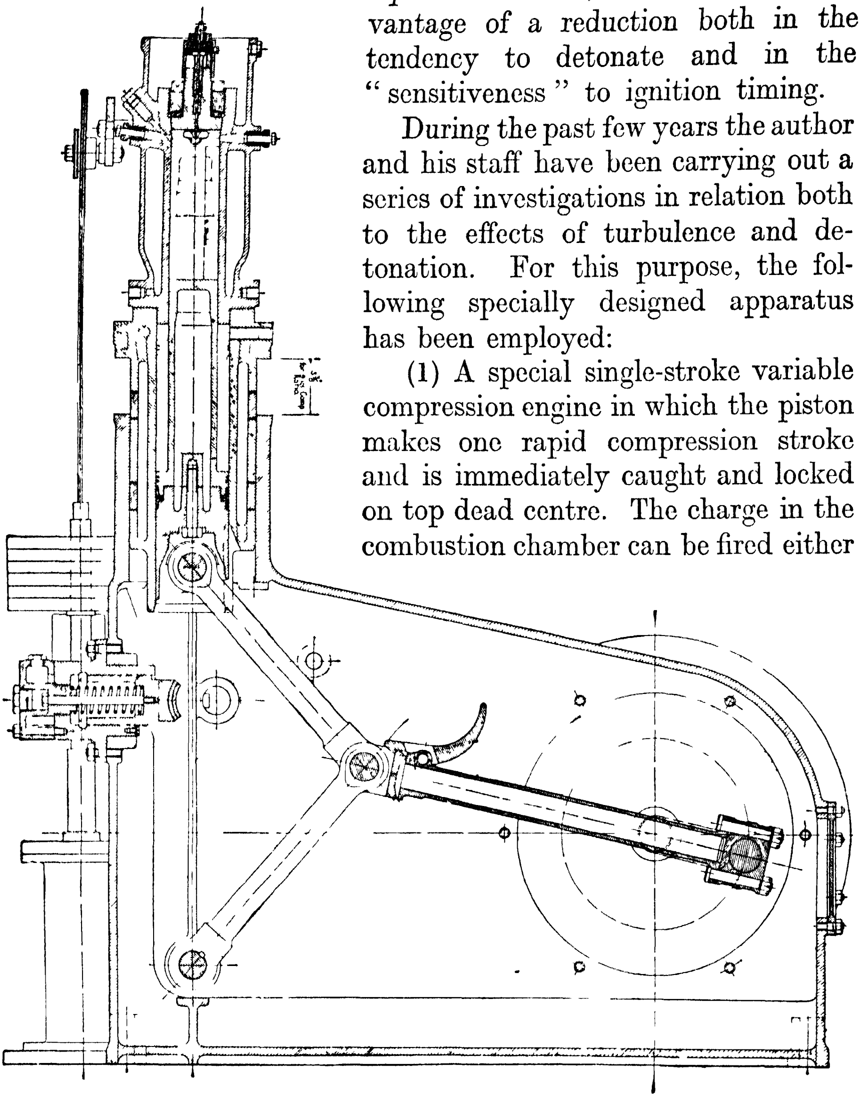
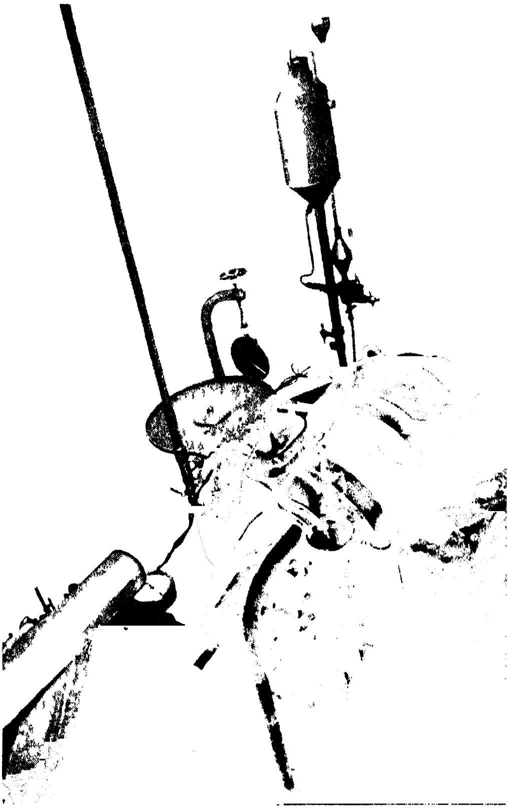


Fig. 51

(2) A special side-valve research engine (figs. 52 and 53) equipped

with very delicate means for measuring the pressure during combustion at each degree of crank-angle, and with a series of quartz windows controlled by a stroboscope for observing the rate of spread of flame outwards from the sparking plug to the farthest point in the combustion chamber. The engine is equipped with readily detachable heads and over eighty different forms and shapes



have been tested and examined as to power output and efficiency, rate of pressure rise, sparking plug position, speed of flame travel, tendency to detonate and all such relevant factors.

(3) A variable compression engine in which also various combinations and positions of both valves and sparking plugs can be tested.

(4) A sleeve-valve engine in which the turbulence can be controlled both in direction and extent by means of guide vanes immediately outside the inlet ports, and recorded by means of

a form of anemometer fitted within the combustion chamber.

(5) A considerable number of research engines built originally for other investigations, but applicable for the purpose of such experiments. In all cases a Dobbie McInnes "Farnboro" electric indicator was employed to determine and record the pressure changes in the cylinder and more particularly the rate of pressure rise during the progress of combustion.

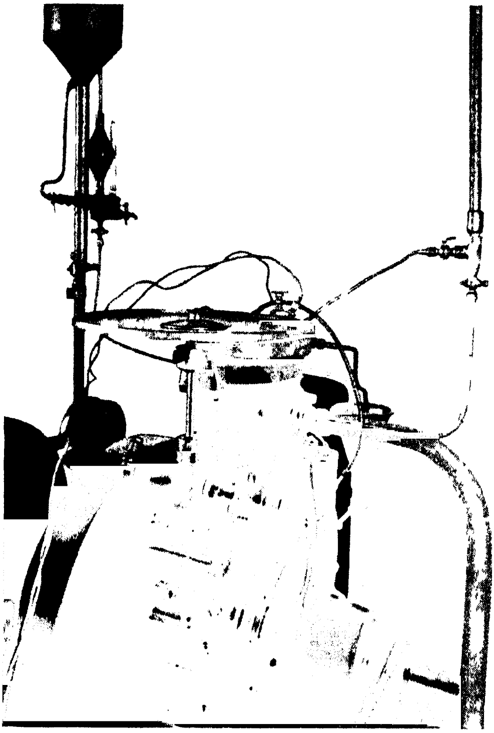


Fig. 53

Such an investigation naturally takes many years and is excessively tedious, for the most meticulous care must obviously be taken to ensure that all conditions are strictly comparable, especially such factors as mixture strength, air, water, and oil temperatures, &c. Moreover, it is obvious that the experimental apparatus must be so designed and constructed as to ensure the highest possible consistency from a mechanical point of view.

The general gist of the experiments carried out has, on the whole, confirmed the conclusions previously expressed, but it is now possible to separate out and to evaluate the various individual factors.

Briefly, the more recent experiments have shown:

(1) *That the rate of burning depends primarily upon the degree of turbulence, and may be expressed in terms of increase of pressure per degree of crank-angle.*

(2) That, to a secondary extent, the rate of burning depends upon the compression ratio; the higher the compression the more rapid the burning and therefore the less need for turbulence. The diagrams, fig. 55, taken consecutively from the variable compression engine, described in Chapter I, at ratios of 4, 5 and 6·0 : 1, but with exactly the same degree of turbulence in each case, show this very clearly.

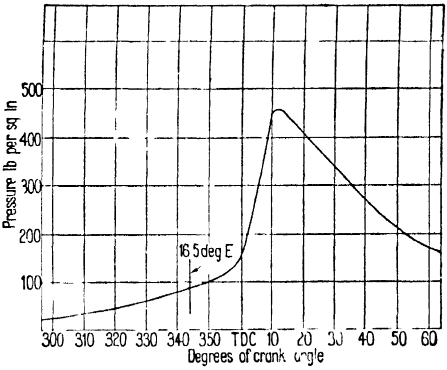


Fig. 54.—Max. press., 466 lb per sq. in. Pressure rise, 34·4 lb. per sq. in. per deg. Ignition advance 16·5 deg. E.

(3) That, at any compression ratio, the best power output and efficiency are obtained on full load when the rate of pressure rise is approximately 30 lb. per square inch per degree of crank-angle, but that at light loads a somewhat greater degree of turbulence is desirable. Fig. 56 shows the observed relationship between power output and rate of pressure rise. It will be seen that over the range

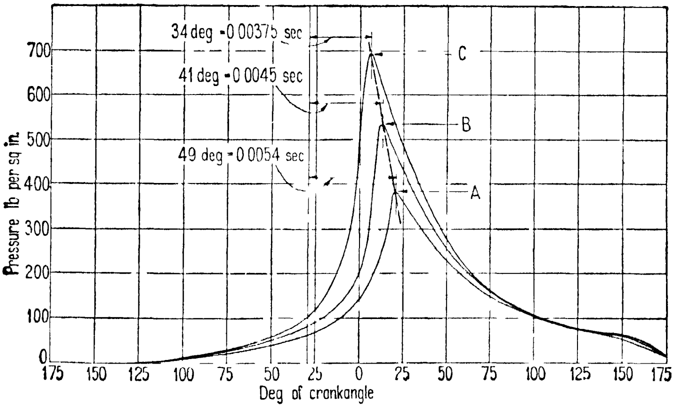


Fig 55—A, R = 4·0 : 1. I·M E P. = 120 lb. per sq. in.
B, R = 5·0 : 1. ,, ,, 136 ,, ,, ,,
C, R = 6·0 : 1. ,, ,, 146·5 ,, ,,

25 lb. to 35 lb. per square inch per degree the curve is fairly flat; below 25 lb. the loss by delayed and incomplete combustion becomes serious; in practice, very serious, for delayed combustion means heavy carbonization, a tendency to burn out or pit exhaust valves, a "sensitiveness" to ignition timing and a tendency to fire back into the carburettor if the mixture is weakened to the economical limit. Above about 35 lb. per square inch the heat loss by intense convection exceeds the gain due to the more rapid burning and the performance once more begins to fall. Also, at these high rates of pressure rise the very abrupt application of pressure intensifies crankshaft spring, piston slap, &c., and renders the engine running very harsh, noisy, and rough.

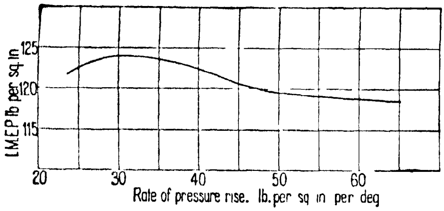


Fig. 56.

(4) That in the so-called "turbulent head", turbulence is set up to an almost equal degree by:

(*a*) The initial velocity through the inlet valve.

(*b*) The velocity through the restricted throat during compression.

It is intensified also by the final ejection, just prior to ignition, of the gases entrapped between the piston and cylinder head.

During the series of experiments referred to, each of the above factors was varied individually step by step, and its influence evaluated.

(5) That though each of the factors (*a*) and (*b*) influences the degree of turbulence to an almost like extent, it is usually undesirable, because of secondary considerations, unduly to intensify either of them at the expense of the other; that is to say, it does not pay, on the grounds of volumetric efficiency, to provide a very small inlet valve and a large throat, thus relying for turbulence mainly on intake velocity, nor, on the other hand, does it pay to provide a very large valve and rely for turbulence on a very restricted throat; for either method alone or when exaggerated unduly tends to restrict the breathing capacity of the engine.

(6) That turbulence from each of the three sources increases proportionately with increase of speed so that the relationship of rate of pressure rise to crank-angle remains substantially constant throughout the whole speed range of the engine; thus a combina-

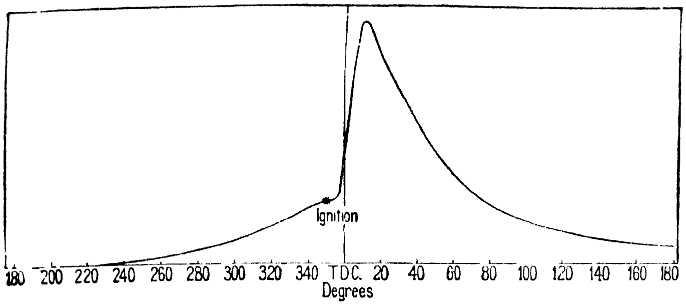


Fig. 57.—Pressure rise 32·5 lb. per deg. Ignition 10 deg. E. Max. press. 438 lb. Speed 800. Full load. O.V.S. Detonation. Max. power mixture strength. I.M.E.P. 122 lb.

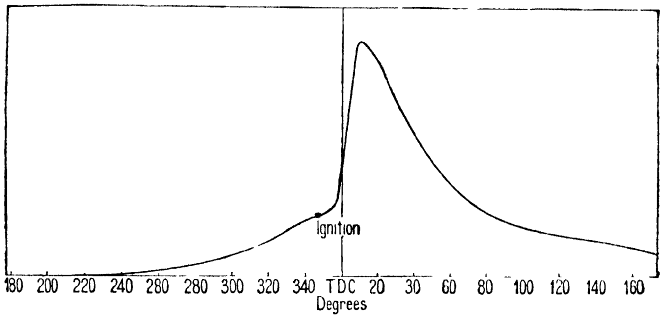


Fig. 58—Pressure rise 32·5 lb per deg. Ignition 13 deg E. Max. press. 446 lb. Speed 1200. Full load. No detonation. Max. power mixture strength. I.M.E.P. 127 lb.

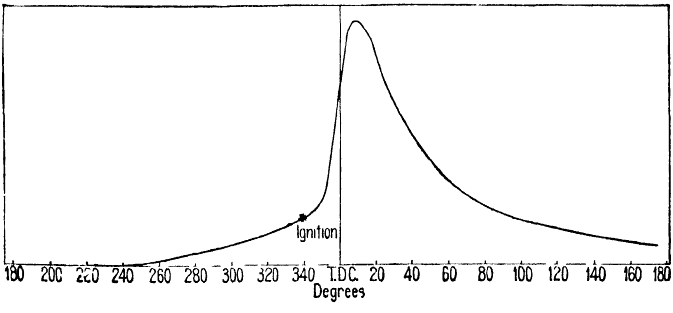


Fig. 59.—Pressure rise 31·8 lb. per deg. Ignition 21 deg. E. Max. press. 442 lb. Speed 1800. Full load. No detonation. Max. power mixture strength.

tion of valve size and combustion chamber proportions designed to give a rate of pressure rise of 30 lb. per square inch per degree at *full throttle will give this rate at any engine speed. A typical example of this is shown by the series of indicator diagrams taken from a* commercial vehicle engine equipped with such a head. (Figs. 57, 58, and 59.)

(7) That in all cases the best possible position for the sparking plug is nearly in the centre of the "effective" combustion chamber, not quite in the centre, but with a slight bias towards the exhaust valve head, since this neighbourhood is necessarily the hottest zone and should therefore have the shortest flame travel. Only a slight bias is desirable however. If pushed too far over towards the exhaust valve the length of flame travel in the other direction then becomes too great.

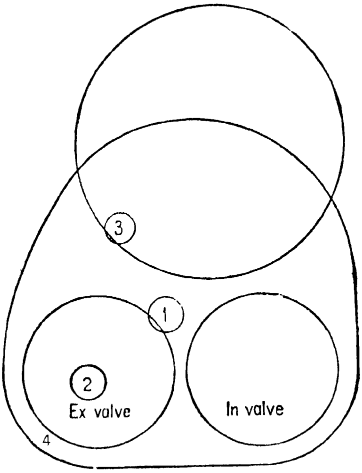


Fig. 60 —Sparking Plug Positions No. 1, Normal position in Ricardo Turbulent Head. No. 2, Over Exhaust Valve.

Experiments on a turbulent head to investigate the effects of various positions of the sparking plug (see fig. 60) gave the following results, shown in Table I.

TABLE I

Position of Sparking Plug.	Full Throttle Performance corrected to C.R. 5:1.		Ignition Advance, degrees Early.	H. U. C. Ratio on Shell No 1 Petrol.
	B M.E P., lb /sq. in.	Fuel Consumption, Pt/B.H.P./hour.		
1	111·9	0·552	17·6	5·87:1
2	109·7	0·577	19·8	5·65:1
3	110·5	0·577	23·0	5·3 :1
4	109·5	0·579	22.0	5·45:1

It will be noticed that position No. 1 is that giving the best results both in power and efficiency, together with the least tendency to detonate. Although in position No. 2 the sparking plug is well over the exhaust valve, a hot region, the tendency to detonate is greater than for the more central position No. 1. The reason for this lies in the greater length of flame travel in position No. 2. In Position 4 the effect is still more marked.

(8) That so far as detonation is concerned, the clearance between the piston and cylinder head in the "inoperative" portion is the main determining factor, and is critical. Here detonation is liable to take place, as shown by observation through the quartz windows, so soon as a certain minimum clearance is exceeded. Fig. 61 shows the observed relationship between clearance space and the tendency to detonate, from which it will be seen that a reduction in clearance from 0·15 in. to 0·10 in. alone permits of the compression ratio being raised by no less than 0·85 ratio, viz. from 5·75 to 6·6.

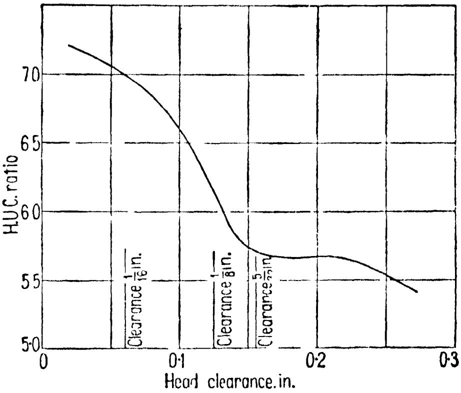


Fig. 61.—Curve showing tendency to detonate plotted against head clearance. Spark in normal position for Ricardo head.

(9) That the effect of turbulence alone upon the tendency to detonate and as distinct from much more important factors such as length of flame travel is appreciable, but not outstanding. It is extremely difficult in any practicable engine to vary the degree of turbulence without varying also other relevant factors which are liable to confuse the issue, so far as detonation is concerned. For the purpose of this part of the investigation the single-stroke engine, with and without the fan running, proved the most instructive. In this apparatus the compression could be varied within wide limits, while the fan speed, and therefore the turbulence, was under complete control. It was found that as the fan speed was increased, so the compression ratio could be raised progressively, when the same fuel was used, at the same mixture strength and the same temperature.

Again, a series of experiments on the research engine No. 2 on which only the area of the throat or passage connecting the combustion chamber proper with the cylinder was varied, showed that, as this area was reduced, and the turbulence and rate of pressure rise increased thereby, so the ratio of compression within the detonation limit could be raised.

Thus a series of tests on a large number of heads all fitted to the same cylinder, and with the same minimum clearance above the piston, but differing only in the throat area, yielded the results shown in Table II.

TABLE II

Degree of Turbulence (expressed in terms of rate of pressure rise, lb. per square inch per degree of crank angle).	Highest Useful Compression Ratio of Shell No. 1 Petrol.
25	5·35:1
30	5·55:1
35	5·80:1
40	6·10:1
45	6·55:1
50	6·85:1
55	7·05:1
60	7·15:1

The above tests may be considered as strictly comparable in every respect except that as the throat area was reduced, so perforce was the maximum length of flame travel, though only to a very small extent. It will be seen that as the turbulence is intensified so is the permissible compression ratio increased, but that over the range practicable with any normal design of engine, namely from 25 to 35 lb. per square inch per degree, the effect of turbulence upon detonation is not of a very large order.

In a similar manner the influence upon detonation of the depth of the layer of gas contained in the clearance between the piston and cylinder head was determined by varying this alone. A series of experiments which could be carried out without the disturbance of any other relevant factor except that decrease of clearance, by increasing the rapidity of ejection of the entrapped gases, tended somewhat to increase the turbulence. The result of these observations taken with five different clearances and with a fixed throat area giving rather high turbulence are given in Table III.

It will be noted that as between heads B and C a difference in clearance of from 0·085 to 0·147 accounts for over one whole ratio of compression, though the observed difference in rate of pressure rise as between these two tests was less than 3 lb. per square inch per degree. The very large difference is here to be accounted for by the fact that a clearance of 0·147 was sufficient to allow detonation to occur in the shallow pocket so formed. In other words the farther portion of the combustion chamber over the piston has, at this larger clearance, become " operative " so far as detonation is concerned. This was confirmed by observation through the stroboscope and quartz windows when the flame could be seen to pass across the clearance space at full velocity, whereas with the smaller clearance space it slowed down very appreciably after entry. Thus, as between heads B and C the effective length of flame travel is increased by 2½ in., i.e. the length of the clearance space lying between the piston and cylinder head.

The effect on turbulence of varying the size of the inlet valve was almost exactly similar to that of varying the restricted throat, that is to say, when the area of the inlet was doubled, the rate of pressure rise was approximately halved and vice versa, thus showing that turbulence can be promoted to an almost like extent by restriction either at the inlet valve throat or in the communicating passage.

Table III

Head.	Clearance between Piston and Head. Inches.	Highest Useful Compression Ratio of Shell No. 1 Petrol.
A	0·022	6·9 . 1
B	0·085	6·8 : 1
C	0·147	5·67 : 1
D	0·210	5·48 : 1
E	0·272	5·37 : 1

A series of tests was carried out in order to determine the influence of shape generally. Keeping the clearance above the piston and the throat area constant, various contours ranging from a plain rectangular section to rounded sides with a " streamline " slope over the piston were also tried several years ago. The effect on performance was negligible, even between the two extremes; the difference in power output, fuel consumption, detonation and all other relevant factors was observed to be less than 1 per cent.

Limitations of the Turbulent Head.—The chief limitations of the ordinary turbulent head are:

(1) That with high degrees of turbulence, although the maximum pressure is not increased to any appreciable extent, the very abrupt application and rise of pressure following ignition causes " spring " of the moving parts, especially the crankshaft, and so gives rise

to a high periodicity vibration accompanied by an indeterminate *drumming noise, the combination of the two being usually described as "roughness". In practice, this roughness determines and limits the degree of turbulence which may be employed. Conversely the degree of turbulence which may be used without roughness is* a measure of the rigidity and stiffness of any engine, more particularly as regards its crankshaft. A really well-designed engine with an adequately supported crankshaft will run quite smoothly with a rate of pressure rise up to 30 lb. per square inch per degree, but only an exceptionally "stiff" engine will stand 35 lb. per square inch per degree without roughness.

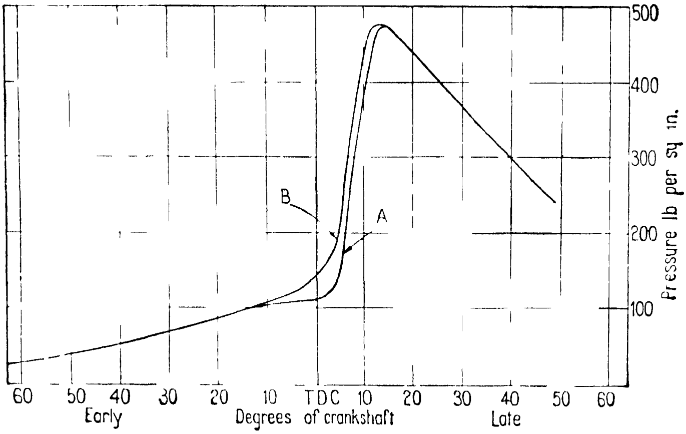


Fig. 62

An examination of the problem of roughness has, therefore, played a very prominent part in the general investigation.

It has been found that roughness is due, not to the absolute rate of pressure rise but rather to the very abrupt application of pressure, and that, if the pressure is applied at first gradually, a very rapid rate of pressure rise may be employed later without any trace of roughness, much in the same manner as a cam-operated valve, even with a very high rate of acceleration, can be made quite silent in operation by the provision of a gradual approach curve to take up all clearance and backlash. In order to provide a gradual initial pressure rise and thus gently to take up all spring and backlash, means were sought whereby a small proportion of the total charge could be isolated, and burned in a more or less stagnant condition, while the main bulk was in a state of normal turbulence. By igniting first this stagnant portion, a slow but gradually increasing rate of pressure rise could be attained initially, followed by extremely rapid burning so soon as the flame reached the main portion of the charge. By isolating and stagnating 15 per cent of the total charge, it was found possible to burn 85 per cent of the charge at a rate of pressure rise even as high as 50 lb. per square inch per degree without any appreciable roughness and certainly with markedly less roughness than in an ordinary turbulent head at a rate of 30 lb. per square inch per degree. The two superimposed indicator diagrams in fig. 62 illustrate the different conditions; in one case the whole charge was burnt at the rate of about 50 lb. per square inch per degree; in the other, about 15 per cent was burnt slowly and the remainder at the rate of about 50 lb. per square inch per degree. In the former case, the running was intolerably rough; in the latter, quite reasonably smooth.

The cylinder head shown in fig. 63, patented recently by the author, and known as the "Shock-Absorber" type shows the practical application of this principle. In this type of head, a shallow flat-roofed pocket is provided with the sparking plug fitted centrally therein. The pocket opens out into the main combustion chamber through a wide but shallow mouth extending about half-way round the circumference, and having a very abrupt rectangular opening. This pocket may be placed anywhere in the cylinder head, but it is convenient to place it over either the inlet or exhaust valve. The capacity of the pocket may be varied at will, depending upon the amount of gradual "take up" required. In practice, the best compromise is reached when the capacity of the pocket ranges between 10 and 25 per cent of the total capacity of the combustion chamber. The gas driven into this pocket during the compression stroke is broken up by the abrupt and sharp-edged

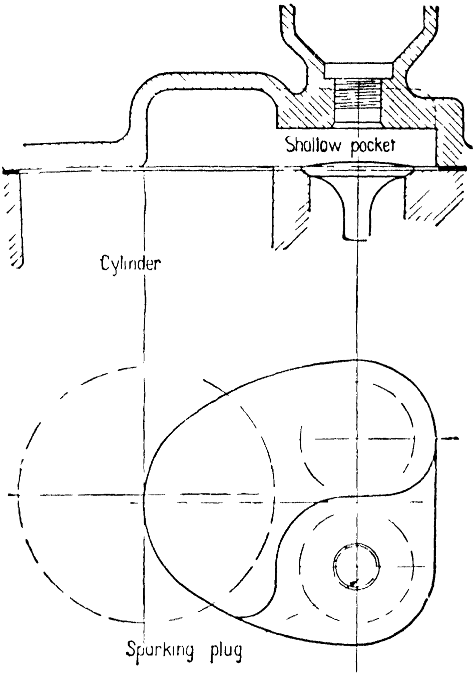


Fig 63.—"Shock Absorber" head

entry into local eddies which die down very soon in so restricted a space.

On ignition, the gas in the pocket burns slowly, but at a steadily increasing velocity, until the flame reaches the open mouth when, owing to the abrupt change of section, the flame front is broken up and disseminated rapidly throughout the main body of the combustion space; stroboscopic and pressure rise observations on one experimental head, in which the capacity of the pocket was approximately 15 per cent of the total volume of the combustion chamber, showed that the time from the actual passage of the spark until the flame reached the open mouth amounted to 30° when the shallow pocket was placed over the inlet valve, while the completion of combustion (up to the point of maximum pressure) of the main body of the charge amounted only to 12°. In other words, the burning of 15 per cent of the total charge occupied 30° of the crankshaft, while that of the remaining 85 per cent occupied only 12°. When the pocket was placed over the exhaust valve, the rate of burning therein was faster by about 4 crankshaft degrees, due to the higher temperature both of the gas and of its surroundings when in this position. See fig. 64.

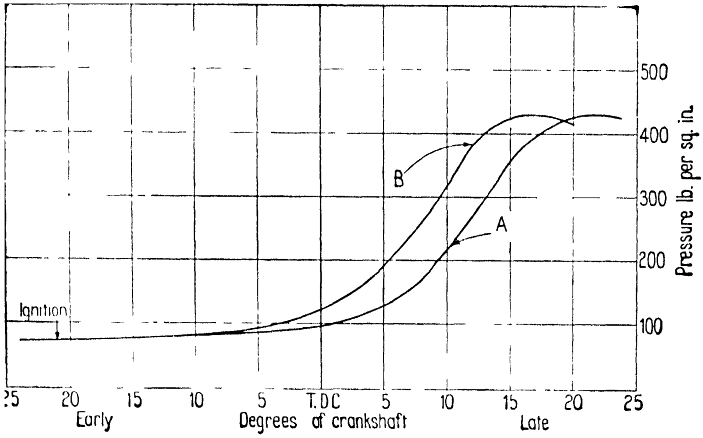


Fig. 64

From the point of view of detonation, although the sparking plug is no longer centrally situated, the "Shock-Absorber" type of head is very slightly better than the ordinary turbulent type, because, so far as detonation is concerned, the effective length of flame travel is little more than that from the open mouth of the pocket to the farthest point in the combustion chamber.

It is found with the " Shock Absorber " type head that the disturbance caused by the burning of the pilot charge and the emergence of the flame on a wide front, tend to create additional turbulence in the main body of the combustion chamber, so much so that when, to a given turbulent head, having a normal rate of pressure rise of 35 lb. per degree, a shallow pocket was applied, the rise of pressure rose to over 45 lb. per degree, though none of the controlling factors were altered. This means, in effect, that for a " Shock-Absorber " type head, the throat connecting the combustion chamber with the cylinder may be enlarged considerably and still allow of the same degree of effective turbulence and the same rate of pressure rise, as with the ordinary turbulent head, in fact, the restriction, as such, becomes less of a necessity.

In either the side valve or in the conventional overhead valve type of engine, the valves are placed in a single row, along the engine, consequently the total available valve area, or in other words, the breathing capacity, is limited by the length of the engine or rather that of the cylinder block.

In practice, the requirements of the crankshaft and its bearings dictate the minimum cylinder centres and therefore the length of the cylinder block. For any given size of cylinder the total available valve area obtainable when all the valves are in a single row and minimum cylinder centres are adhered to, is such as to limit the peak of the power curve to a piston speed of about 3000 ft. per minute, and this limitation applies alike to either side valve or overhead valve engines so long as the valves are in a single row. In order, therefore, to obtain the peak of the power curve at piston speeds well in excess of 3000 ft. per minute, it becomes necessary to arrange the valves in two rows. In racing-car engines it has now become the invariable practice to employ two rows of inclined overhead valves, each row operated by a separate overhead camshaft, thus allowing of the use of a centrally placed sparking plug between the two camshafts. This provides probably the most efficient form of head possible with any arrangement of poppet valves, but is open to the objection that it is complicated, costly, and inaccessible; for this reason it is seldom used except for purely racing purposes. The object of the " High Power " head is to reproduce as nearly as possible the above conditions, without resort to an overhead camshaft, and in a form which is relatively inexpensive and accessible.

The high-speed capacity and the high efficiency of the twin overhead camshaft engine are due to:

(1) The ample breathing capacity afforded by having the inlet *and exhaust valves in separate rows, thus doubling the length available for valve accommodation.*

(2) The central position of the sparking plug.

(3) The very short length of flame travel due to the central sparking plug and the absence of any corners or pockets where the gases may be entrapped and detonated, thus allowing of the use of a high ratio of compression.

(4) The low weight of the moving masses between the cams and the actual valve heads, thus allowing of high rates of acceleration and therefore of a very efficient valve opening diagram.

In the "High Power" type of head, these conditions are reproduced as far as possible in the following manner:

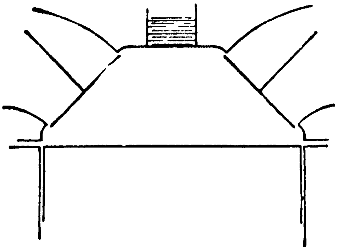


Fig. 65

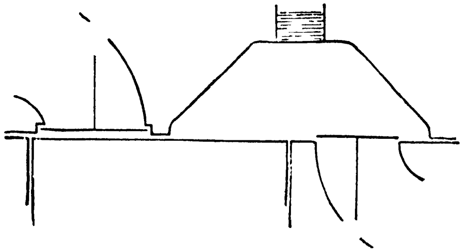


Fig. 66

(1) A double row of valves is obtained by placing the inlet valve overhead and operating it through a push rod and rocker; the exhaust valve, however, is retained in its usual position, as in a side-valve engine. In this there is, of course, nothing in the least novel.

(2) The sparking plug is placed in the centre, or almost in the centre, of the effective combustion space.

(3) The overhead inlet valve is situated in the flat "inoperative" portion of the combustion chamber, the latter is offset as in the ordinary turbulent head and embraces only the exhaust valve and a small proportion of the piston area; thus, the chamber, in plan, is similar to but considerably more compact than in the case of the ordinary turbulent head, since it embraces one valve only, and the length of flame travel is no greater than in the twin overhead camshaft type of engine. The whole combustion chamber, in fact, is very nearly similar to that of the racing engine, but is offset sideways in relation to the piston, as shown in figs. 65 and 66. The conditions as regards turbulence are practically the same as in the

ordinary turbulent head, and, of course, a "Shock-Absorber" pocket can be employed if desired. The location of the inlet valve *entirely* outside the operative portion of the combustion chamber is, of course, both the novel and the essential feature of this type of head.

(4) Since the inlet valve opens directly into the cylinder barrel and beyond the combustion chamber, there is no resistance in series with it due to the restricted throat, and conversely there is no need to limit the restriction, i.e. the degree of turbulence, to avoid throttling. Again, since we do not depend for turbulence on the entering velocity through the inlet valve, this valve can be made of almost unlimited size; moreover, since it is outside the combustion chamber, its size does not influence either the area of the chamber or the length of flame travel.

(5) In the case of the exhaust valve the mass interposed between the cam and the head is little or no greater than in the overhead camshaft type. In the case of the inlet valve it is, of course, considerably greater, but the inlet valve, unlike the exhaust, can be masked to any desired extent, and an efficient opening diagram can be obtained thereby with a relatively low rate of acceleration and therefore allow of a very high engine speed. Again, the inlet valve is not opened against a high back pressure so that the operating mechanism can be made considerably lighter than would be the case if the exhaust valve were placed overhead. Also, since the overhead valve is the cool one, no large clearance is required to allow for expansion.

Samples of this head were tested on the No. 2 research engine; these, at the same compression ratio and the same speed, gave approximately 7 per cent more power and 5 per cent lower fuel consumption than the best of the ordinary turbulent heads. Using a throat area which would give a rate of pressure rise of 40 lb. per square inch per degree in the ordinary turbulent head, the rate in the "High Power" head was only 29 lb., showing that the removal of the inlet valve to beyond the combustion chamber had reduced the turbulence. With this type of head, brake mean pressures as high as 122 lb. per square inch have been maintained up to well over 4000 R.P.M. on ordinary commercial petrol.

The high-power type of head has been adopted and is in use in several very high-speed engines on the Continent, where it has given results almost identical with those obtained from the conventional twin overhead camshaft type. Owing to its high inherent efficiency and to the mechanical design, which allows of very ample water circulation round the exhaust valve seating and valve stem, it is particularly well adapted to supercharging; all the more so, since the short flame travel allows of the use of a fairly high ratio of compression even when supercharged.

Determination of the Combustion Chamber Efficiency.—It has been shown that the efficiency of the combustion chamber is the most important of all the factors which control both the power output and the thermal efficiency of an engine. Unfortunately, however, it is, in most cases, very difficult to determine even the relative efficiency of two different designs of combustion chamber, in the case of multiple-cylinder engines, because irregularities in distribution play so important a part as to render a determination based on the known consumption of fuel per horse-power hour of very little value. It is very commonly found that of two engines the measured fuel consumption per horse-power hour is the same, despite the fact that one may have a considerably more efficient combustion chamber, and therefore develop more power than the other. At first, it would appear that the design of both engines is equally efficient, whereas in fact the former is much the more efficient of the two, but the practical value of its high inherent efficiency has been annulled by greater irregularities in distribution. Were it practicable to change the whole distribution system from one engine to the other a very marked change in efficiency would at once be observed; but this in practice is seldom, if ever, possible, because in almost all existing designs of multi-cylinder engines a considerable proportion of the induction passage is embodied in the cylinder casting.

The most practical method which the authors have yet found for determining the true efficiency of a multi-cylindered engine independently of all irregularities in distribution, defective carburation, &c., depends upon the fact that in the case of any hydro-carbon fuel, or alcohol, the heat energy liberated by the combustion of any given weight of air is almost exactly the same, however much excess of fuel may be present, provided always that there is an excess of fuel, or, in other words, so long as the mixture is over-rich the thermal efficiency, based on that portion which is burnt, will be the same over a very wide range of mixture strength on the rich side, with the result that over the range from 5 per cent rich to 35 per cent rich the heat liberated by unit weight of air is constant to within extraordinarily narrow limits.

If, now, instead of measuring the weight of fuel consumed, we measure the air consumption and calculate the efficiency obtained in terms of lb. of air consumed per horse-power hour, we shall arrive at a means of determining the true thermal efficiency of an internal-combustion engine irrespective of any losses due to irregularities in distribution, defective carburation, precipitation of liquid fuel, etc. If, by air measurement, we find that the thermal efficiency of an engine is low, then we know that the combustion chamber design is at fault, and that no amount of juggling with the carburettor or distribution system will avail.

On the other hand, as so often happens, an engine apparently well designed and with what would appear to be an efficient form of combustion chamber shows on test a very poor fuel economy; in such a case, we can, by ascertaining the air consumption, determine at once whether the fault lies in the engine design or in that of its distribution system. Given reliable means for measuring accurately the air consumption, it is necessary only to set the carburettor to give a rich mixture, in practice to set it to the mixture giving maximum power (which on petrol is about 15 to 20 per cent rich) and take readings of air consumption: all that is needed being to ensure that the mixture strength is such that the weakest cylinder in the group is receiving a mixture not less than about 5 per cent rich. Since there is an extremely wide range available beyond this, i.e. up to about 40 per cent rich (after which combustion becomes delayed again as on the weak side), this presents no difficulty whatever.

The absolute value found for the thermal efficiency by this method is, in all cases, somewhat higher than that obtained from the fuel-consumption readings when working with the most economical mixture strength, and this is so even in the case of single-cylinder engines, where distribution losses can be almost entirely eliminated, the difference being due to:

(1) The larger increase in specific volume when an over-rich mixture is used.

(2) The slight loss due to condensation of liquid fuel on the cylinder walls which escapes combustion and ultimately finds its way past the piston into the crankcase.

(3) The small loss due to precipitation of liquid fuel on the walls of the induction pipe; this fuel enters the cylinder in gulps of liquid which are not completely vaporized, at any period during the cycle.

(4) The small loss of liquid or vapour due to "blow-back," or

reverse flow in the induction pipe due to the sudden closing of the inlet valve.

Once the air consumption is known, the thermal efficiency of any engine can be determined from the formula :

$$E = \left(\frac{\text{I.H.P.}}{\text{lb. of air per hour}}\right) \times C,$$

where C is a constant representing the amount of heat liberated by the combination of 1 lb. of air ; for all petrols it may be taken as 196.

The following table gives the effective calorific value, the mixture strength for complete combustion in terms of air/fuel ratio by weight, the amount of heat liberated by the combination of 1 lb. of air, and the value of C in the formula above, for a representative selection of fuels, from which it will be seen that over the whole range of available fuels the heat liberated by the combination of 1 lb. of air is substantially the same.

The figures given in the fourth column are calculated as for a

TABLE IV

Fuel.	Effective Lower Calorific Value B.Th.U.s per lb.	Air Fuel Ratio.	Heat liberated by 1 lb. of Air B.Th.U.s.	Value of Constant.
Petrol samples				
(1)	19,200	15·05	1275	197·0
(2)	19,020	14·7	1295	195·0
(3)	19,120	14·8	1293	195·5
(4)	18,900	14·6	1295	195·0
(5)	19,090	14·9	1282	197·0
(6)	19,250	15·0	1285	196·5
(7)	18,920	14·7	1288	196·0
Kerosene	19,100	15·0	1275	197·0
Hexane	19,390	15·2	1275	197·0
Heptane	19,420	15·1	1285	196·0
Benzene	17,160	13·2	1320	192·5
Toluene	17,660	13·4	1315	193·0
Cyclohexane ...	18,910	14·7	1290	196·0
Heptylene... ...	19,320	14·7	1320	192·5
Ether	16,830	13·0	1295	195·0
Ethyl Alcohol—				
99 per cent ...	11,950	8·95	1333	190·0
95 „ ...	11,125	8·4	1330	190·5

mixture giving complete combustion, as also the value of the constant C; but over a very wide range of mixture on the rich side, the variation is, in all cases, very small indeed, and appears to be almost exactly the same in the case of all volatile liquid fuels, so that the possible error due to variations in mixture strength is extremely small.

The following tables, Nos. V to VII, give the results of typical

TABLE V

Petrol R = 5 : 1, Sample No. 4

Date of test, 5/7/21

Mixture Strength.	Lb. of Air per Hour.	Indicated Mean Pressure, lb per sq. in.	Indicated Horse-Power.	Lb. of Air per I H P. Hour.	Indicated Thermal Efficiency.
					Per cent.
Correct	196·0	132·0	32·0	6·13	32·1
Plus 5 % excess fuel	196·5	135·0	32·7	6·0	32·8
,, 10 ,,	197·0	136·5	33·1	5·95	33·4
,, 15 ,,	197·5	137·5	33·3	5·93	33·2
,, 20 ,,	198·1	138·0	33·4	5·94	33·15
,, 25 ,,	198·8	138·0	33·4	5·96	33·05
,, 30 ,,	199·5	137·5	33·3	5·98	32 85
,, 35 ,,	200·0	136·5	33·4	6·01	32·7

Maximum thermal efficiency calculated from fuel consumption 32·1 per cent, with mixture 16 per cent weak.

TABLE VI

Ethyl Alcohol 99 per cent R = 5 : 1

Date of test, 27/8/21

Mixture Strength.	Lb. of Air per Hour	Indicated Mean Pressure, lb. per sq. in	Indicated Horse-Power	Lb. of Air per I.H.P. Hour.	Indicated Thermal Efficiency.
					Per cent.
Correct	199·0	141·0	34·2	5·82	32·9
Plus 5 excess fuel	199·5	143·0	34·6	5·77	33·3
,, 10 ,,	200·0	144·5	35·0	5·72	33·65
,, 15 ,,	201·0	145·5	35·25	5·70	33·7
,, 20 ,,	202·0	146·5	35·5	5·69	33·75
,, 25 ,,	203·0	147·0	35·6	5·70	33·7
,, 30 ,,	204·0	147·3	35·7	5·72	33·65
,, 35 ,,	205·0	147·6	35·8	5·73	33·6

Maximum thermal efficiency calculated from fuel consumption 33·0 per cent, with mixture 15 per cent weak.

TABLE VII

Ethyl Alcohol 95 per cent R=5 : 1

Date of test, 16/8/21

Mixture Strength.	Lb. of Air per Hour.	Indicated Mean Pressure, lb. per sq. in.	Indicated Horse-Power.	Lb. of Air per I.H.P. Hour.	Indicated Thermal Efficiency.
					Per cent.
Correct	203·0	145·0	35·1	5·79	33·1
Plus 5 % excess fuel	203·5	146·7	35·5	5·72	33·6
„ 10 „	204·0	148·0	35·8	5·71	33·65
„ 15 „	205·0	149·0	36·0	5·70	33·7
„ 20 „	206·0	149·5	36·15	5·70	33·7
„ 25 „	207·0	150·0	36·25	5·70	33·7
„ 30 „	208·0	150·3	36·3	5·71	33·65
„ 35 „	209·0	150·5	36·35	5·74	33·4

Maximum thermal efficiency calculated from fuel consumption 32·9 per cent, with mixture 15 per cent weak.

tests on petrol and alcohol (the latter both nearly pure and 95 per cent) over a wide range of mixture strength from correct to 35 per cent over-rich, taken on the variable compression engine described in Chapter II, while at the foot of each table the maximum thermal efficiency as deduced from the fuel consumption at the most economical mixture strength is given for purposes of comparison.

In Table VIII are given the results of air measurement tests at varying compression ratio, ranging from 4·0 : 1 to 7·0 : 1, the fuel used in this case being benzol. Here again the agreement between the calculated thermal efficiency from the air consumption with a rich mixture, and from the fuel consumption with a weak mixture, is on the whole very consistent.

Similar tests carried out on a six-cylinder aero-engine with a compression ratio of 4·7 : 1 showed an air consumption of 6·4 lb. of air per I.H.P. hour over a range of mixture strength from 10 per cent to 25 per cent over-rich, using standard aviation petrol. The air efficiency in this case works out at 31 per cent, a figure very considerably greater than that obtained from the fuel consumption, which was only 28·2 per cent at the most economical mixture strength. The discrepancy in this case represents the loss due to irregularities in distribution, etc. A comparison of these results with those given previously for the variable compression research engine is very interesting, all the more so because both engines have

TABLE VIII

Varying Compression Ratio

Fuel, Benzol, about 20 per cent rich

Date of test, 16/10/21

Compression Ratio.	Lb. of Air per Hour.	Indicated Mean Pressure, lb. per sq. in.	Indicated Horse-Power.	Lb. of Air per I.H.P. Hour.	Indicated Thermal Efficiency as found by Air Measurement.	Indicated Thermal Efficiency as found by Fuel Measurement 15 per cent weak.
					Per cent.	Per cent.
4	203	125·0	30·3	6·70	28·8	27·7
5	195	136·5	33·1	5·90	32·8	32·0
6	189	145·0	35·2	5·37	35·9	35·0
7	185	152·0	36·8	5·03	38·3	37·3

nearly identical forms of combustion chamber, very nearly the same cylinder capacity, and, in both cases, the charge is ignited by sparking plugs at opposite sides of the combustion chamber. That the aero-engine does not show, by the air-consumption test, so high an efficiency as the research engine at the same compression ratio, namely, only 31 per cent as against 31·7 per cent, is to be explained probably by :

(1) Although two plugs were used they were sparked by two separate magnetos, and therefore not so accurately synchronized.

(2) The aero-engine had a relatively shorter stroke, so that the combustion chamber was flatter, and therefore somewhat less efficient.

(3) The mechanical condition of the aero-engine was probably not so good, i.e. there was probably more leakage loss.

The difference in fuel consumption was, however, much more marked, for the maximum thermal efficiency was 30·9 per cent reckoned on the fuel consumption at a compression ratio of 4·7 : 1 in the case of the single-cylinder research engine and only 28·2 per cent in the six-cylinder aero-engine. After allowing for the different combustion-chamber efficiencies of the two engines, the loss by irregularities in distribution, loss of unburnt fuel, etc., of the aero-engine is about 10 per cent, while in the single-cylinder it is only about 2·5 per cent. The indicated mean pressure was found to be exactly the same in both engines, namely, 133 lb. per square inch ; but measurements of volumetric efficiency showed that, while the research engine had a volumetric efficiency at this compression ratio of 76·2 per cent, that of the aero-engine was 78 per cent, a difference which very approximately compensates for the lower combustion chamber efficiency of the latter engine. Again, tests carried out on a four-cylinder commercial vehicle engine with a compression ratio of 4·24 : 1 showed an air consumption of 7·75 lb. per I.H.P. hour = 25·6 per cent thermal efficiency as against 6·58 = 30·4 per cent in the variable compression engine at the same ratio, from which it may be deduced that the efficiency of the combustion chamber was only 85 per cent that of the research engine at the same comparison ratio. In this particular engine the design of the cylinder head was very defective, the valves being placed in deep-set shallow valve pockets with the sparking plugs directly over the inlet valves. The fuel efficiency at the most economical mixture strength was, however, about 23·8 per cent, showing that the efficiency of distribution, if such a term may be used, was, in the case of this four-cylinder engine, as high as 93 per cent. In other words, this engine made up, to some extent, so far as fuel efficiency was concerned, for bad cylinder design by having a quite unusually efficient distribution system, but the bad cylinder design showed itself in the low power output obtainable, the indicated mean effective pressure being only 93 lb. per square inch as against 129 lb. per square inch in the variable compression engine under exactly similar conditions. Had the volumetric efficiency been the same in both cases, the mean effective pressure would have been $\frac{85}{100} \times 129$, or about 110 lb. per square inch; that it was, in fact, only 93 lb. was due again to defective cylinder design, whereby the free entry of the gases after leaving the inlet valve was obstructed by the surrounding walls of the shallow valve pockets. Air measurements showed also that, while the volumetric efficiency of the research engine at this compression ratio, and at the same temperature, was 77 per cent, that of the four-cylinder engine was only 66 per cent, so that the maximum M.E.P. should have been $\frac{66}{77} \times 110$, or 94 lb. per square inch, a figure which agrees very closely with the 93 lb. actually measured during the tests. This latter is rather a striking example of an inherently defective engine, which showed a comparatively good economy because its induction system was unusually efficient.

In the absence of any means for measuring the air consumption, a fair estimate of the efficiency of different forms of combustion chamber can be gained by comparing the maximum mean effective pressure, but this again assumes that the volumetric efficiency is the same in both instances.

Such an assumption is, of course, not always justifiable, but it is at least fair to assume that the variations in volumetric efficiency as between two somewhat similar types of engine will be very much less, and will have a much smaller influence on the determination of the combustion-chamber efficiency, than the variations in mixture strength, as between individual cylinders, despite the rather exceptional example quoted above. Where means are available for air measurement, the efficiency of any form of combustion chamber can very readily be determined from the measured air consumption.

If an engine consumes its air efficiently, then it is an efficient engine, and to render it economical in fuel is a question solely of carburation and distribution. If its air consumption is heavy, then no amount of finessing with carburettor adjustments or distribution design will render it efficient.

Methods of Measuring Air Consumption.—Until recently there were two main methods of measuring air consumption. In the first place, there is the air bell, the device employed for many of the tests here recorded, which gives an accurate determination, but has the serious drawback that with its pipework, &c., it constitutes a plant of considerable size, and is therefore an inconvenient and in no sense a portable apparatus. Secondly, there are the "kinetic" meters, examples of which are the orifice and venturi types. These are simple and handy to use, but hopelessly inaccurate owing to "root mean square" errors, unless the flow is quite steady. Somewhat similar errors have to be contended with in the hot wire and Callendar types of meters. In all cases such errors can be reduced by means of a smoothing capacity, but often the capacity required is of quite impracticable size.[1]

Viscous Flow Air Meter.—Thanks to the ingenuity of J. F. Alcock a compact, easily fitted piece of apparatus has been developed by means of which, with very little trouble, air consumption measurements may be taken to within plus or minus 1½ per cent. This instrument can deal with pulsating flow and does not require the assistance of a large smoothing capacity in series. It can therefore be used to the same purpose on either single or multi-cylinder engines.

The meter has proved to be invaluable in the author's laboratories, in which it was developed, for measuring the scavenge air consumption of high-speed two-stroke research units and the air consumed by highly-boosted units, as well as for complete calibrations of engines of more moderate output.

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[1] For fuller discussion of the question see "Pulsating Air Flow Measurements," N. P. Bailey, *Proc. A. S. M. E.*

The Alcock Viscous Flow Air Meter consists essentially of a honeycomb of long narrow triangular passages, 3 in. long and 0·017 in. in height, of cross-sectional area 0·00017 sq. in., through which, within the working range of the meter, the air flow is viscous. The resistance is therefore proportional to the velocity, and the " root mean square " error of the kinetic type of meter—such as the sharp-orifice or the venturi—is eliminated. The resistance is measured by manometer across the " meter element ".

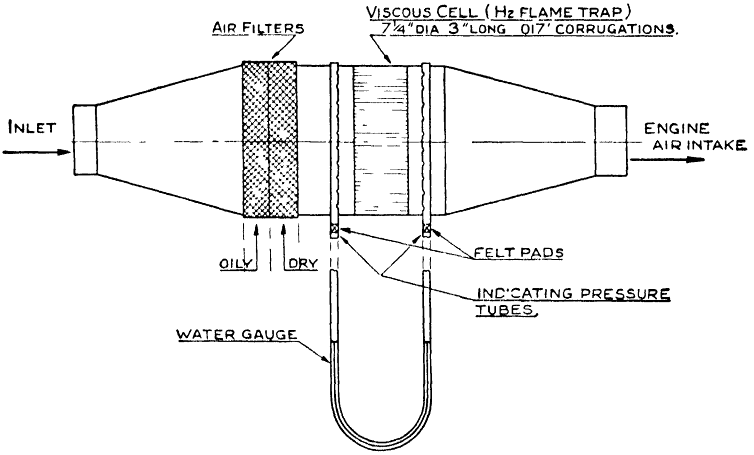


Fig. 67.—Alcock's Viscous Flow Meter

Moreover, by special design of the manometer connections, errors due to the flow in and out of the connections which is set up by pressure variation in the line are avoided.

A diagrammatic sketch of the meter is shown in fig. 67. The air passes through a cleaner and then through the meter element, which is constructed of alternate layers of flat and corrugated sheet metal wound round a core. The manometer connections are tubes spanning the face of the element and having a single row of holes in each, those in the upstream tube pointing in a downstream direction and those in the downstream tube pointing upstream. This provides a reverse kinetic head which automatically corrects the small kinetic pressure loss in the element itself due to entry effects. In the connections themselves felt pads are provided which give a condition of viscous flow in the tubes, thus eliminating " manometer connection " errors. The manometer head was found after many experiments to be proportional to the gas velocity, and even with the pulsating flow present in the induction pipe of a single-cylinder engine the accuracy of measurement was unimpaired. The meter is not, of course, an absolute standard, and has to be calibrated by comparison under steady flow conditions with a " German Standard " nozzle which itself has been checked against a gas-holder.

A useful feature of the instrument is that since the pressure difference is proportional to the rate of flow, the range of air quantity measurable with reasonable accuracy is much greater than with kinetic types in which the pressure varies as the square of the rate of flow.

CHAPTER V

LUBRICATION AND BEARING WEAR

Three factors have to be considered in the design of any bearing: first, the ability to carry the necessary duty in the space available with a reasonable margin of safety against breakdown; next, the rate of wear of the bearing surfaces; and, finally, the energy lost in friction.

Piston friction will be dealt with in a later chapter, so that the bearings only will be considered here.

These are generally of the "plain" or sliding friction type. Ball and roller bearings, being of a fundamentally different nature, are not included.

Where two surfaces, apparently in contact, are moving relatively one to the other, there are three possible cases to be considered.

In the first case, that of "dry" friction, the surfaces are in actual contact without any lubricant. In this case the friction is very great, and only very low loads and speeds can be imposed without seizure. This case never occurs when a bearing is functioning properly, so need not be dealt with further.

The second condition, that of "greasy" friction, occurs when the surfaces, though virtually in contact, are actually lubricated with some substance which discourages their mutual adhesion. The lubricant in such cases appears to function by exerting some kind of chemical action on the metallic surfaces.

Lastly, there is "viscous" friction, in which the surfaces are completely separated by a film of lubricant: this is clearly by far the most desirable state of affairs, and, fortunately, it is one which can easily be attained in a well-designed and adequately lubricated bearing, and may, in fact, be regarded as the normal condition.

In a journal bearing the necessary oil film between the loaded surfaces is maintained by a wedging action, due to the fact that the shaft sets itself eccentrically in its journal. This is made clear in fig. 68, where it can be seen that the oil in the wide space A is dragged by the rotation of the shaft into the narrow space B, thus forcing the surfaces apart. Under these circumstances the frictional loss and the thickness of the oil film for any given conditions can be evaluated theoretically, and it has been found that the results so obtained hold good in practice. These two factors—film thickness and friction—depend, for any given bearing, solely on the load, the speed, and the viscosity of the lubricant.

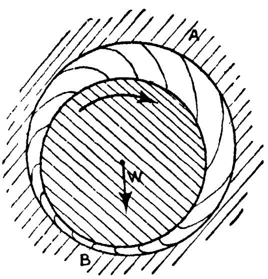


Fig. 68

The influence of these three factors is roughly as follows: Increase of load, *alteris aequis*, increases the friction, though not nearly in direct proportion, and decreases the film thickness. Increase of speed increases both the friction and the film thickness, as does also an increase in the viscosity of the lubricant. However, increase of either load or speed, by increasing the rate of energy loss, heats up the oil, and thus decreases its viscosity.

This fact, in the case of an increase of load, to some extent nullifies the increased friction; it still further decreases the film thickness, while with increased speed the theoretically increased film thickness is actually reversed. Thus both increased load and speed tend to reduce the film thickness.

It is clear that, as no surfaces are perfectly smooth, there is a limiting thickness of oil film at which the high spots of the two surfaces begin to bridge across the oil film.

At these points the oil film is no longer of appreciable dimensions, so that the laws of " greasy friction " begin to apply. The frictional force, under these circumstances, is, at such points, considerably higher than with viscous friction, and obeys totally different laws. The chief factor appears to be a property, probably of a chemical nature, called " oiliness," which tends to reduce friction between two surfaces in contact. It is quite distinct from viscosity, as quite thin oils, such as sperm or rape, can have it in a far greater degree than more viscous substances, such as treacle, from which it appears to be entirely absent. Unfortunately the data on this subject is very limited and largely contradictory, but it is fairly well established that oils of animal or vegetable origin, such as sperm, rape, and castor oils, are considerably " oilier " than the hydrocarbon mineral oils, while viscous substances of a " sticky " nature are entirely devoid of the property.

It is clear that if the lubricant has but little " oiliness," the friction at the " high spots " will be exceedingly high, and that the heat so generated will still further reduce the thickness of the oil film, thereby making matters worse, and so starting a " vicious circle " ending in seizure, unless the bearing lining has so low a melting-point that it can fuse locally without boiling off the lubricant as in the case of white-metal.

With an oily lubricant, on the other hand, the increase in friction at the " high spots " will be much less, and the risk of seizure correspondingly deferred.

The influence of " oiliness " is thus only of importance where, owing to excessive loading, unsuitably disposed surfaces, or inadequate oil supply, a complete film of oil cannot be maintained. It has apparently no influence on the friction under normal conditions, though it may enable one to use, in any given case, a thinner oil, with correspondingly reduced friction, by relying, as a reserve in case of abnormal conditions, upon oiliness rather than upon excess viscosity.

To return to the consideration of normal conditions, the maintenance of an adequate oil film necessitates efficient arrangements both for the removal of the heat generated and for the continuous replenishment of the oil in the bearing.

The heat is got rid of, to some extent, by the oil which forms the actual oil film, but mostly either by conduction from the bearing surfaces, or by excess oil which runs through or over the bearing without actually forming the load-carrying film. The advantages of forced lubrication are mainly due to the cooling effect of the excess lubricant.

The supply of oil to the bearing is greatly facilitated by the " pumping " action caused by alternating loads, it being found that any given bearing will carry an alternating load considerably in excess of the maximum steady unidirectional load. Again, a narrow bearing loses a far greater proportion of its oil through side leakage than does a wide one, so is correspondingly less efficient when subject to " natural " lubrication, while with forced lubrication a narrow bearing will carry a heavier continuous load per square inch of projected area because of the greater quantity of oil which can be circulated through it and therefore the better cooling. It is also necessary that the surfaces should be of such a shape that the wedging action, which maintains the oil film at the highly loaded parts of the bearing, can take place, for the pressure which must exist in order to carry the load is far in excess of any that a forced lubrication system can supply.

Fortunately, journal bearings are naturally suitable, as explained earlier, but with thrust collars special measures have to be taken, as in the Michel bearing. However, in practically all cases, in so far as high-speed internal-combustion engines are concerned, such thrust loads as are involved are most conveniently dealt with by the adoption of ball thrust races.

When conditions are too severe for even a molecular oil film to survive, the surfaces come into actual metallic contact with one another—that is to say, when the oil film becomes so thin that the two surfaces come within the zone of molecular attraction, an exceedingly high temperature is at once set up, resulting in local fusion of the surfaces. In some cases such local fusion may, by removing a high place in the surface of the bearing, relieve the loading at this particular point and so permit of the re-formation of the oil film. Unless this occurs, and the oil film re-forms immediately, the surface fusion will spread until the bearing either seizes solid or the bearing metal melts and runs out. When both bearing surfaces are hard and of anything approaching the same melting point, the surface fusion is generally accompanied by partial welding, and the two surfaces become inextricably locked. When the material forming one surface is relatively soft and has a low melting point, as in the case of white metal, the bearing merely melts, either locally or completely; in the former case, as already pointed out, local melting may be due to the existence of a high spot, and the fusion of this high spot may at once relieve the cause of trouble and permit of the restoration of the oil film in time to prevent any further spread of surface fusion: this is possible because the melting point of white metal is well below the boiling point of the lubricant. Local fusion is a very common occurrence in white metal lined bearings, especially when new and not too well fitted, and is often quite harmless.

The intensity of heat flow when an oil film breaks down locally is very surprising; it is quite common to find two case-hardened steel surfaces fused together locally and the temper of the surfaces undisturbed within less than $\frac{1}{2}$ in. from the point of fusion. This corresponds to a temperature difference of something like 2500° F. in $\frac{1}{2}$ in.

The case of bearings submitted to very heavy loads and low rubbing speeds is somewhat different. Here the time element enters prominently into the question. Since at low speeds the wedging action, tending to maintain the oil film, is very slow, the effect of the pumping action, due to change of load, is of correspondingly increased importance. There is abundant evidence that under severe loads and favourable conditions even bronze bearings may be crushed without breaking down the oil film. So long as facilities are available for the replenishment of the oil, and so long as the rubbing velocity is such that the product of loading and rubbing velocity does not exceed a certain figure—that is to say, so long as the heat generated does not exceed the rate at which it can be dissipated by replenishment with cool oil or by conduction—a journal bearing will not fail from pressure. Failures of heavily loaded, slow-moving journal bearings are almost invariably due either to an interruption in the oil supply or more frequently to bending or distortion of one of the members, causing excessive local pressure and heat flow. The only bearings in a high-speed internal-combustion engine submitted to very heavy loading with low rubbing velocity are the gudgeon-pin bearings. Experience has shown that when the gudgeon-pin is supported in such a manner that it does not distort appreciably through bending, maximum pressures up to 6000 lb. per square inch may quite safely be carried, without forced lubrication and without perceptible wear.

Wear of Bearings and Shafts.—In view of the fact that most bearings are completely oil-borne, it is perhaps a little surprising at first sight that wear should take place at all, since the two surfaces never actually come into contact. The explanation appears to be that all wear is due to the abrasive action of small particles of grit carried by the oil. These particles, which are, for the most part, so small that they cannot be removed by filtration, are carried by the oil into the bearing and there embed themselves in the softer of the two surfaces; thus in a white metal lined bearing the particles of grit invariably embed themselves in the soft white metal. Once partially embedded they proceed to lap the shaft. It is clear that only those particles of grit which project far enough from the softer material to span the oil film and so actually touch the other member can cause wear. Other things being equal, therefore, the rate of wear depends upon the thickness of the oil film, which, in turn, depends upon the pressure and temperature—the cooler the lubricant, or the lighter the pressure, the greater is the thickness of the oil film and, therefore, the greater the distance across which the particles of grit must reach before they can come in contact with the harder member.

Again, the rate of wear depends upon the hardness of the surfaces of the materials. It is common knowledge that when a soft-steel shaft runs in white-metal bearings it is the shaft and not the bearing material which wears; this is perfectly natural, since the particles of grit will always tend to embed themselves in the softer surface of the two and so proceed to cut or lap the other and harder surface. In order to reduce wear it is desirable that the difference in hardness between the two surfaces should be as large as possible: thus in the case of a steel shaft running in white-metal bearings, the softer the white metal the more readily will the particles of grit embed themselves entirely out of harm's way, while the harder the shaft the less readily will it submit to the cutting action of such particles of grit as are not completely embedded.

All available evidence indicates that the once popular idea that a hard white metal should be used, and that its surface should be rendered even harder by hammering or driving a taper mandrel through it, is quite erroneous—the surface of the white metal should be as soft as is consistent with the necessary resistance to crushing. In the case of two very hard surfaces, such as case-hardened steel against cast-iron or hardened steel, very little wear occurs, the probable explanation being that the hardness of both surfaces exceeds that of the particles of grit, so that the latter are merely ground up between the two surfaces and do not get any opportunity of embedding themselves in either or of cutting them.

It has been observed that such bearing surfaces, even when freely exposed to road dust, etc., do not wear readily, but that when carborundum is introduced along with the lubricant very rapid wear takes place.

It is when two surfaces, both relatively soft and of somewhat similar hardness, are employed that the most rapid wear takes place. A soft-steel shaft running in bronze bearings wears very rapidly indeed, unless the load factor is so low as to permit of a very thick oil film being maintained.

One very striking example of excessive wear between two surfaces of nearly similar hardness is to be found in the case of copper aluminium alloys and phosphor-bronze. An alloy consisting of 88 per cent aluminium and 12 per cent copper affords an excellent bearing material for hardened-steel shafts. It is light, which is often important, is an excellent conductor of heat, and is readily cast and machined. When, however, a phosphor-bronze shaft is run in bearings of this aluminium alloy, the shaft, which is slightly the harder of the two, wears away with almost incredible rapidity. Similarly, experiments which the author has carried out with bronze piston rings in an aluminium cylinder and with an aluminium piston in a bronze-lined cylinder resulted, in the former case in the rings wearing down to half their original thickness in eight hours, and in the case of the bronze-lined cylinder the liner wore about 0·010 in. oval in a run of twelve hours. In neither case did the aluminium show any appreciable wear.

Again, a soft-steel shaft, running in a copper-aluminium bearing, wears away very rapidly.

Probably the worst possible results are to be found when two similar and relatively soft materials are used for the two members of a bearing, for then not only is the difference in surface hardness reduced to zero, but the opportunities for welding together in the event of a failure of the oil film and consequent local fusion are at a maximum.

In the case of cast-iron or hardened steel, both surfaces are so hard as to be very little affected by grit, but, in the event of a breakdown of the oil film, the two are very liable to become welded.

Experience with aluminium pistons has shown :

(1) That when these are fitted in soft-steel cylinders the cylinder bore wears very rapidly ;

(2) When fitted in hard-steel cylinders—0·4 carbon—the wear is very slight ;

(3) When fitted in cast-iron cylinders finished by grinding, wear of the cylinder bore takes place if the grinding material has not been thoroughly removed. Such wear does not take place when the cylinders are reamed or when lapped after grinding.

Maximum Pressures on Bearings.—So far as high-speed bearings are concerned—that is to say, when the rubbing velocity exceeds about 8 ft. per second—the load factor only, that is, the product of load and speed, need be taken into account; the maximum pressure, so long as it is not high enough to distort or crush the bearing material, is of little moment, since it is not applied for a long enough period to have any influence on the conditions of lubrication.

Limiting Load Factor.—The highest load factor which can safely be carried by a bearing depends upon :

(1) The system of lubrication, whether forced, natural, or fed with a measured quantity of fresh oil.

(2) The viscosity of the oil.

(3) The facilities available for conducting away the heat generated in the bearing.

Under the conditions ruling in high-speed engines in which the circulation of oil is limited to that with which the piston scraper rings can cope, with forced lubrication, and good facilities for dispersing the heat, load factors as high as 30,000 lb. ft. per second can be safely carried in a journal bearing with alternating load. With such a load factor the rate of wear is of course considerable, but there are many examples of crankshaft centre bearings in aero-engines with load factors as high and even higher than this, but about 12,000 to 15,000 lb. ft. per second appears to be the upper limit of sustained load for a splash- or trough-lubricated bearing.

Where very high rubbing velocities are involved, much higher load factors can be carried when floating bushes are employed. Such bushes, floating freely between the two members, rotate at an intermediate speed, so that the rubbing velocity between either face is halved; also they permit of a much greater circulation of cooling oil through them. Under such conditions the load factor may be increased by 50 per cent without imperilling the bearing.

Maximum Load.—Where the rubbing velocity is low the only limit to the maximum load is set by the rigidity of the members. There is no danger of the oil film being broken down by pressure alone, provided there is scope for natural replenishment. When serious distortion takes place, the load factor may be increased locally—that is to say, the pressure may all be concentrated on one point in the bearing; and since the rubbing velocity is the same at all points, it follows that the product of pressure and rubbing speed may be excessive at one point, causing rapid local heat flow and ultimate breakdown of the oil film.

Load Factor and Wear.—Other things being equal, the rate of wear may be taken as being almost directly proportional to the load factor. It depends, however, very largely on the facilities for cooling; thus, in the case of crankshaft bearings, for equal load factors with forced lubrication the rate of wear on the main journal bearings is more rapid than on the crankpin bearings, because the crankpin bearings are always better served and therefore cooled with oil, since it is flung out to them by centrifugal force.

Oscillating Bearings.—When the motion is oscillating only, the wear on the members is no longer uniform; this objection can, however, usually be overcome by allowing the harder member to float freely—thus, in the case of a gudgeon pin, if this is fixed, either to the piston or the connecting-rod, local wear will take place, but if allowed to float freely in bearings, both in the connecting-rod and piston, local wear on the pin can be avoided; further, a much heavier load can be carried, because the rubbing velocity between any of the members is halved.

From the above considerations, it is clear that, other things being equal:

(1) The friction of a bearing, when freely lubricated, is nearly proportional to the load factor on the bearing, and depends, though to a lesser extent, upon the nature of the surfaces—the smoother the surface the lower the friction.

(2) The rate of wear is also proportional to the load factor.

(3) When oil of higher viscosity is used, the friction is greatly increased at first; but on account of the greater amount of energy dissipated in shearing the oil film, the heat flow is greater, the temperature is therefore higher, with the result that, after running some time, the reduction in viscosity due to the higher temperature nearly compensates for the higher initial viscosity, and so the conditions as regards friction and the thickness of the oil film ultimately become nearly similar. They do not become quite similar, because owing to the higher temperature of the bearing the rate of dissipation by radiation and conduction is greater, consequently the bearing never reaches so high a temperature relative to the viscosity of the oil as when a thinner oil is used.

Types of Oils.—The oils used in internal-combustion engines fall into two main divisions—mineral oils and those of animal or vegetable origin. "Compounded" oils, and mixtures of the two types, are also used.

Mineral oils, which are composed of various hydrocarbons, mostly of the paraffin series, are by far the most frequently used. This is partly because of their lower cost, and also on account of their chemical stability, which renders them less prone either to carbonization or to oxidation or gumming. On the other hand, they do not appear to possess the property of "oiliness" to the same extent as vegetable or animal oils. Except in cases where bearings are very severely loaded and near their limit, mineral oils are probably the most suitable, and so far as the author has been able to determine by experiment there is no advantage to be gained from the use of vegetable or compounded oils in any well designed and properly lubricated engine.

For ball or roller bearings a pure mineral oil would appear preferable, since such an oil is less liable to form corrosive acids in service.

Animal oils, such as whale and lard oils, and vegetable oils, such as rape or castor oils, are largely composed of the esters of fatty acids. Their chief virtue lies in their high "oiliness," which is of use in cases where the oil supply is necessarily limited, as in crankcase compression two-stroke engines, or where severe local over-loads, due to distortion, etc., are probable. Their defects lie in their comparative instability, which renders them liable to become gummy and acid by exposure to the air, and also causes them to carbonize more rapidly than mineral oils. They are also expensive, and, the supply being necessarily limited, would become more so if their employment became general. Their use, therefore, should be, and generally is, limited to exceptionally high-duty engines, and a few other special cases.

Carburation

The function of the carburettor is not, as is so often supposed, to gasify the fuel, but rather to provide constant proportions of finely divided liquid fuel and air under all conditions of speed or load. The gasification or vaporization of the liquid fuel takes place in part in the induction system, and in part in the cylinder of the engine.

The requirements of a good carburettor are that it shall—

(1) Provide a constant predetermined ratio of fuel and air at all speeds and at all loads, under constant conditions.

(2) Pulverize the fuel as finely as possible under all conditions.

(3) That when the throttle is opened suddenly it shall provide, momentarily, an over-rich mixture, for reasons which will be explained later.

(4) Provide an over-rich mixture for starting or running idle at very slow speeds.

(5) Be provided with automatic or at least readily controllable means of enriching the mixture throughout the whole or at least the lower part of the range, until the carburettor and induction system are fully warmed up.

(6) Be simple to adjust in the first instance and unlikely to get out of adjustment in use.

Probably no carburettor yet made conforms to all these conditions, though they are not impossible of compliance.

It is worth while to examine each of these conditions separately and to see what their compliance involves.

The first consideration, namely, that of providing a uniform mixture strength under all conditions, or of "metering," as it is generally termed, is the basic problem in carburettor design. The simplest possible expression of a carburettor is a jet to which liquid fuel is supplied at a constant level, such jet being situated in the centre of a venturi nozzle, or choke-tube, through which the whole of the air passes on its way to the engine. The depression in the choke-tube is therefore at all times a function of the I.H.P. of the engine, and this depression is relied upon to draw petrol from the jet. Unfortunately, the laws governing the flow of liquid from a jet, and of air through a venturi throat, are not the same, for the one medium is a liquid and the other is a gas. As the speed at which the air flows through the choke-tube increases, so the flow of fuel also increases, but at a considerably greater rate, with the result that, if the proportionate sizes of jet and choke-tube diameters are chosen to give a "correct" mixture at any one speed, the mixture will be too weak at a lower speed and too rich at a higher speed, as shown approximately in fig. 69.

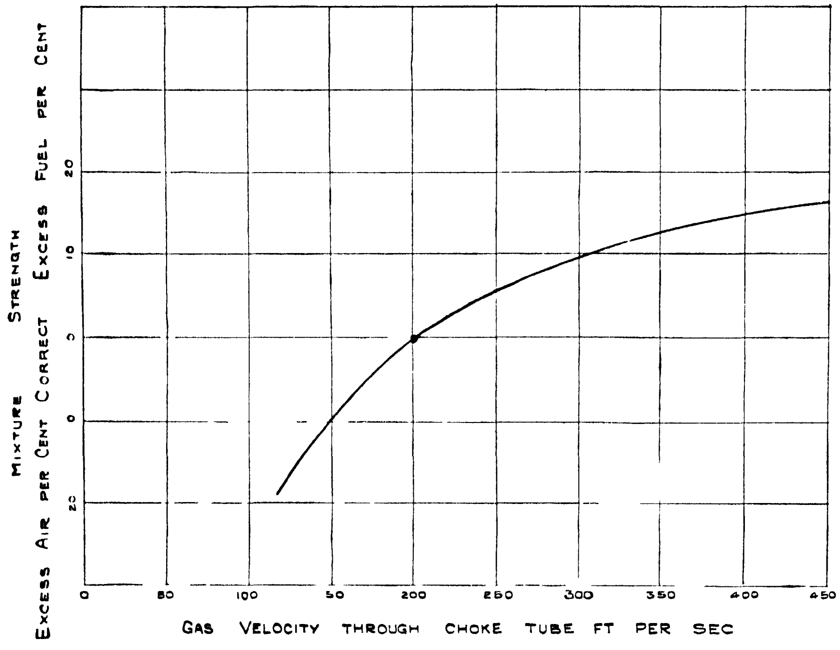


Fig. 69

To this simple form of carburettor some means of compensation must therefore be provided, and for most purposes it must be automatic. There are numerous methods of compensating, but they may be divided broadly into two main groups:

(1) In which means are provided for supplying, automatically, additional air as the power output increases.

(2) In which means are provided for supplying automatically additional fuel as the power output decreases.

Intermediate between these groups are methods providing means for checking the flow of fuel through the jet by obstructing it by means of a reversed air-flow, etc.

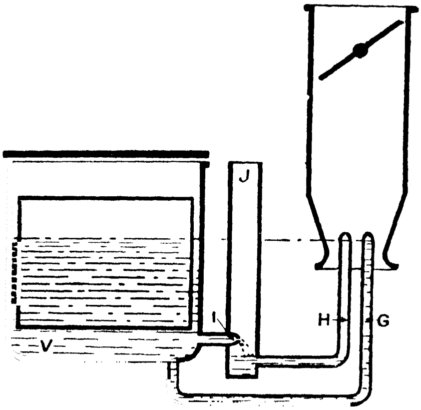


Fig. 70.—Diagrammatic Arrangement, Zenith Carburettor

The first group includes all such devices as automatic extra air valves operated by suction. Broadly speaking, these are not very satisfactory, because they involve the addition of a constantly moving part, which cannot readily be lubricated, and unless the movement of this part is controlled by an efficient dash-pot serious wear is liable to occur; on the other hand, if it is controlled by a dash-pot, then its movement will be somewhat sluggish, though this is not necessarily a disadvantage in view of the third requirement stated previously. In any case, however, it is always desirable to avoid the use of an additional moving part if possible.

The second group includes those carburettors in which compensation is effected by means of an additional jet fed by gravity from the float chamber and open to atmosphere; the flow of such a jet is unaffected by the depression in the choke-tube. Carburettors belonging to this group can be adjusted to give fairly accurate metering under all conditions of speed or load; and since they contain no moving parts to wear or possibly to stick, they are, in the author's opinion, to be preferred. The basic principle of this type of carburettor is illustrated in fig. 70, while fig. 71 shows approximately the rate of flow from either jet. It will be seen that, as the power output and therefore the depression in the choke-tube increases, the flow of liquid from the main jet increases rapidly, while that from the gravity-fed jet, which has a constant head of only about one and a half or two inches, remains substantially constant. Its proportional flow therefore decreases. It is obviously possible by a suitable selection of jet sizes to keep the fuel air ratio very nearly constant over a wide range of speed and load.

In addition to these general groups there are large numbers of purely mechanical devices whereby either the fuel supply, the air supply, or in some cases both, are varied mechanically by the movement of the throttle level. Such devices cannot possibly afford true compensation for all conditions of speed or load, since no change can be effected without movement of the throttle. For certain purposes, however, such, for example, as marine work in which the torque and speed vary in a fixed relation, mechanically compensated carburettors are probably quite satisfactory. The advantages of this form are that a mechanically compensated carburettor can be made very cheaply; it has only one jet to look after and no adjustment which can be deranged. It will fulfil the requirements of a marine engine, but certainly will not give accurate metering when applied to engines in which the speed or torque may vary without movement of the throttle.

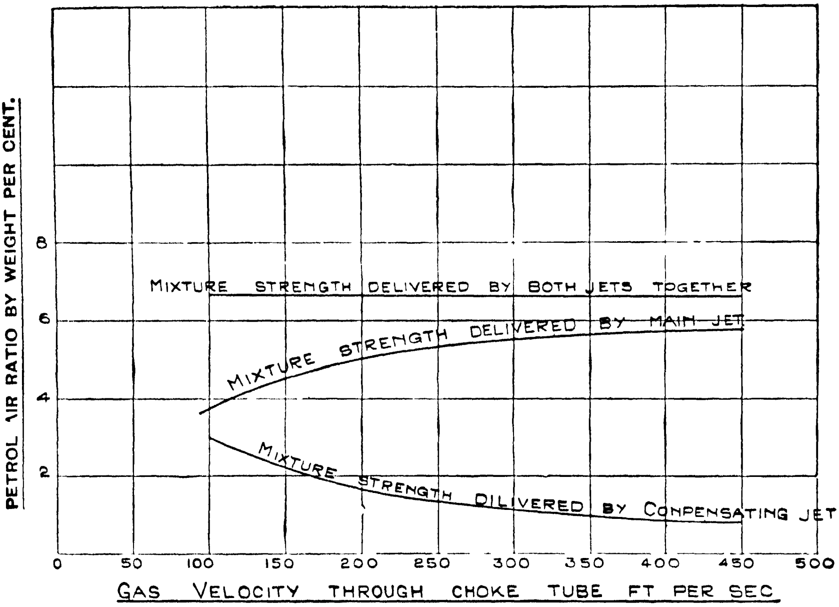


Fig. 71.—Mixture Strength delivered from Main and Compensating Jets, Zenith Type Carburettor.

The second condition, namely, thorough pulverization, is not at all easy to comply with. It is, however, an exceedingly important factor, because it is most desirable from every point of view to keep the suction temperature as low as possible. Whether the fuel enters the cylinder in a liquid or a gaseous state, it will, so long as it is finely divided, be completely evaporated after its entry to the cylinder, on coming in contact with the hot residual products therein.

In a previous chapter it has been shown that it is always very desirable to keep the suction temperature as low as possible, and to this end it is often preferable to allow the fuel to enter in a finely divided but still liquid state, and so to make use of the latent heat of evaporation of the liquid to lower the temperature in the cylinder. This, however, is possible only when the liquid is very finely pulverized, and when the induction system is so arranged that reasonable uniformity of distribution can be obtained without too much differentiation due to the unequal inertia of air and liquid particles; if delivered in coarse drops, these will coalesce and precipitate on the walls of the induction system, finally entering the cylinder in gulps of liquid, which will never evaporate. These pass through the cylinder unburnt, while a proportion of the liquid fuel will find its way past the piston and into the crankcase, where it will contaminate the lubricant.

In most carburettors the velocity of the air past the jet is relied upon to pulverize the fuel, and, for this purpose, every effort is made to obtain a high velocity at the least possible sacrifice in power output.

Unfortunately, however, pulverization becomes of most importance when the engine is running at low speeds or reduced loads, i.e. when the quantity of air passing, and therefore its velocity, are at a minimum. With a venturi orifice of the best possible design the volumetric efficiency, and therefore the power output of the engine, will be penalized severely if the air velocity exceeds 400 ft. per second, and the author has found that to obtain a good compromise between maximum power output on the one hand and good economy on reduced loads on the other, the mean velocity through the choke-tube should not exceed 400 ft. per second when four cylinders are drawing from a single carburettor, 330 ft. per second for three cylinders, and about 250 ft. per second for single cylinders, the lower velocity in the latter cases being permissible because:

(*a*) With less than four cylinders the suction is intermittent and the maximum velocity therefore considerably greater.

(*b*) The fewer the number of cylinders drawing from any one carburettor the shorter the total length of induction pipe.

To obtain better pulverization one or other of two methods may be employed:

(1) A very small choke may be used and the bulk of the air admitted elsewhere, the bulk supply of air being cut off as the load is reduced—this entails a combination of mechanical and fluid compensation.

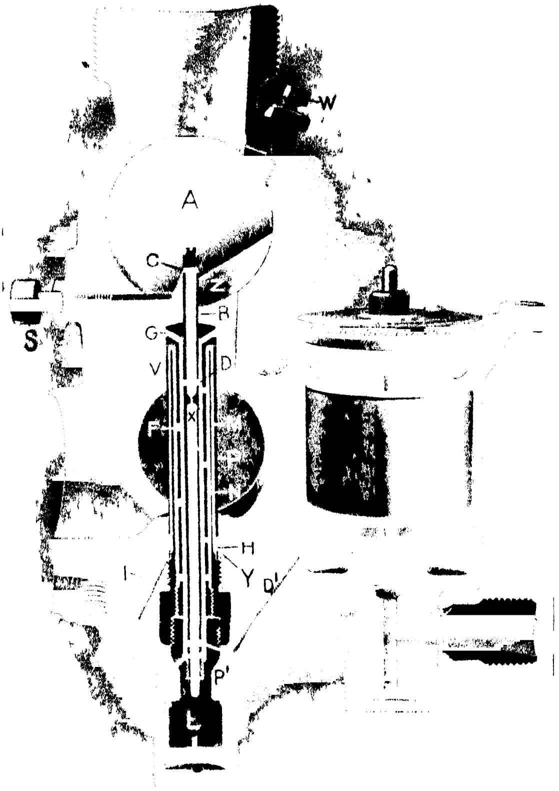


Fig. 72 —Section, Claudel Carburettor

(2) What is termed a shrouded or diffuser jet may be used in which air is drawn through the liquid to form an emulsion, which is then delivered from the jet, as employed in the Claudel carburettor, figs. 72 and 73.

The former has the advantage that it becomes possible at reduced loads to maintain the velocity not only past the jet, but also throughout a considerable proportion of the induction system. It carries with it, however, the disadvantage that the carburettor becomes somewhat complicated.

The diffuser or shrouded jet gives good pulverization at the jet itself, but, owing to the low velocity in the whole of the induction system at light loads or low speeds, particularly the latter, the finely divided particles are given too much opportunity to coalesce. It carries with it, however, the additional advantage that the flow of air through the fuel in the jet tends to effect a certain, though limited, measure of compensation.

For the carburation of all engines liable to sudden demands, such as all road vehicle engines and all engines under the control of a sensitive governor, it is most important that, on the sudden opening of the throttle, the carburettor shall deliver momentarily an over-rich mixture. The reason for this is as follows:

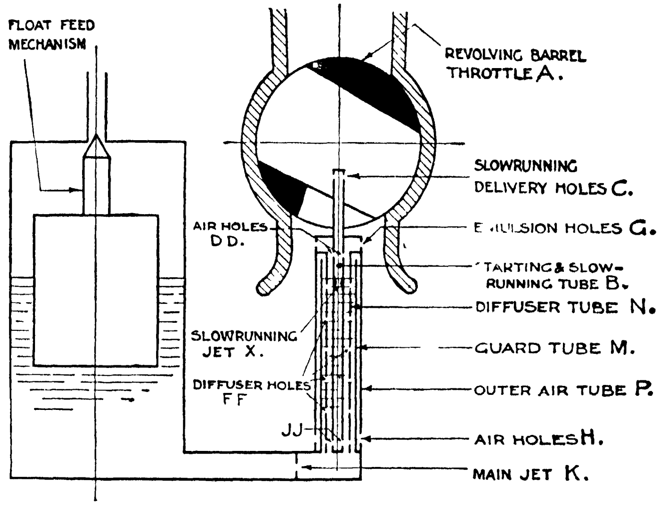


Fig. 73.—Diagrammatic Arrangement, Claudel Carburettor.

When an engine is running light or at a very much reduced load the pressure in the induction system may be only about 5 lb. per square inch absolute. At this pressure and even at quite low temperatures almost the whole of the fuel flowing through the induction system will be evaporated and the walls of the induction passages will be dry. If now the throttle be opened suddenly the pressure in the system will at once rise from, say, 5 lb. per square inch to nearly 15 lb. per square inch absolute, while the temperature conditions will remain unaltered. Now, although the fuel may evaporate completely when under a pressure of only 5 lb. per square inch, when the pressure is raised by nearly three times, this will no longer be the case, unless the induction system be very hot. The immediate result is that the first portion of the fuel admitted after the throttle is opened deposits at once on the walls of the induction system, and, unless the mixture supplied by the carburettor is very rich, that which reaches the cylinders is far too weak to burn; this state of affairs will continue until sufficient fuel has been supplied thoroughly to wet the walls of the induction system, for when working under full throttle conditions the whole of the walls are normally lined with a layer of liquid, the thickness of which depends both on the temperature of the walls themselves and upon the velocity of flow through them. The practical effect of this is that, when the throttle is opened suddenly after idling, the engine will splutter and backfire or even stop firing altogether for a few revolutions; if now the throttle be closed again, the half-formed wet layer will immediately re-evaporate and the engine will run steadily again; by repeatedly opening and closing the throttle the necessary wet layer can be built up gradually. To obviate this difficulty one or other of three expedients must be adopted:

(1) The carburettor must be set to deliver, at all times, an over-rich mixture.

(2) The walls of the induction system must be maintained at so high a temperature that little if any liquid fuel can lie upon them, even at atmospheric density.

(3) The carburettor must be provided with some means whereby a little liquid fuel is stored up when running idle, and delivered to the induction system immediately the throttle is opened.

The first of these may be dismissed at once as altogether too extravagant. The second method can, at best, be but a partial remedy only, though a certain amount of pre-heating is essential in the case of present-day petrols, whose mean volatility is low and most of which have a final boiling point of over 350° F. To raise the induction system, however, to such a temperature as will prevent entirely any condensation even of the highest boiling fractions is practically out of the question, and would, in any event, so reduce the power output, increase the tendency to detonate, and, by raising the whole cycle temperature, so lower the efficiency of the engine, as to be quite outside the range of practical politics.

The third method, namely, the momentary supply of an over-rich mixture, meets the case satisfactorily; it costs nothing in power output, and permits both of working normally with the most economical mixture strength, and of reducing the heat input to the induction system.

In the case of carburettors using a gravity-fed compensating jet, this condition can be met by providing a well having a capacity sufficient to supply a 100 per cent excess of fuel for, say, 3 or 4 cycles, and fed from the compensating jet. When running on full throttle this well is normally dry, but when idling the well fills up to nearly the level in the float chamber. So soon as the throttle is opened the sudden depression caused by the inflow of air to the induction system draws the whole contents of the well into the induction system and provides, momentarily, an over-rich mixture.

When a shrouded or diffuser jet is used the same effect can be brought about by providing in the annular passage around the jet sufficient capacity to meet this instantaneous demand

Both the Zenith and the Claudel carburettors cater for this condition, the former by means of a well fed by gravity, and the latter by the use of a diffuser jet with large capacity. Neither, however, provides sufficient capacity to meet the case of a multi-cylinder engine with a relatively cool induction system, and it has become a common practice in present-day carburettors to incorporate a small plunger pump actuated mechanically by the throttle lever. When the accelerator pedal is depressed, a small plunger positively discharges a quantity of petrol through a small hole into the carburettor tube in the form of a jet. The plunger cylinder is recharged when the accelerator pedal is allowed to rise and the plunger returns to its former position.

The fourth condition, namely, that it shall provide an over-rich mixture for starting, is met, in nearly all cases, by the provision of a separate, or pilot jet, and some provision is usually made whereby this jet drops out of action so soon as the engine has attained sufficient speed to bring the main jets into operation. In those carburettors using a gravity-fed compensating jet, the pilot is fed from the compensating jet and automatically drops out of action so soon as the level in the well falls beyond a certain point.

To start from cold, i.e. 60° F., on ordinary commercial petrol, it is necessary to provide a minimum mixture strength from three to four times over-rich, depending upon the vapour tension of the fuel and upon the actual temperature of the mechanism. For the purpose of starting merely, there is no objection to using an excessively over-rich mixture, provided that its supply is discontinued so soon as the engine is under way. When the pilot jet draws from a well fed by the compensating jet and is but slightly submerged, as in the Zenith or Claudel carburettor, it will deliver an excessively rich mixture only when the level is at a maximum, and this again can occur only when the engine is at rest, for the very minimum running speed will suffice so to lower the level in the pilot jet chamber as to cut this jet either partially or completely out of action. When, however, the pilot jet is fed direct from the float chamber, it is necessary, in order to obtain a sufficiently rich mixture, either to flood the carburettor and so both raise artificially the level in the pilot jet and at the same time expose for evaporation a considerable surface of liquid petrol in and around the air intake to the carburettor, or partially to close the air intake to the carburettor and thereby subject the main jet to excessive suction.

The fifth condition, namely, adjustment of mixture strength to compensate for changes in temperature, is met only in those carburettors which are provided either with variable jets or with a control on the air supply. It is an important condition, but one which is generally unprovided for; indeed, it is not at all an easy one to meet adequately. Although an engine should require the same mixture proportions, once the induction system has been wetted, whether it is hot or cold, yet it is none the less desirable to reduce slightly the size of the jet as the carburettor warms up, because the viscosity of petrol and indeed of most volatile liquid fuels varies considerably with temperature, and, with a given size of jet and a given depression, the quantity of fuel passed will increase as the temperature of the liquid rises, due to the reduced viscosity. The influence of the variation in the viscosity of the liquid can be reduced to the minimum by employing always very short jets or at least a very short length of orifice of small diameter, but it cannot be eliminated entirely by such means.

In all carburettors which are used for aircraft, and therefore required to operate under wide variations both of density and temperature, it is absolutely essential to provide some means of varying bodily the mixture strength. This is effected by means of what are termed altitude controls. Altitude controls take a number of different forms, notably that in which the float chamber is hermetically sealed and the air space in it connected to the carburettor at two points, one below the choke-tube and the other between the choke-tube and the throttle. Between these two points there is a considerable difference in pressure, due to the resistance offered by the choke-tube. A control cock is fitted in the passage connecting above the choke-tube, and this passage is made very much larger, the other being in the nature of a small permanent leak only. When the control cock is closed the pressure in the float chamber is equal to that of the outside air, since it is balanced by the permanent leak, but on opening the control cock the pressure in the float chamber falls to something between that ruling in the passsage above the choke-tube and that of the outside air, depending upon the respective areas of the connecting passage-ways. As the pressure in the float chamber is reduced, so the level in the jet falls, and less liquid is delivered from it. By adjustment, therefore, of the control cock it is possible to lower the level of liquid in the jet to any desired extent, and so to vary the mixture strength. This method and the several variations of it serve admirably for aircraft when the engine is driving a propeller and the torque-speed characteristic is therefore definite throughout the range; but it is in the writer s opinion somewhat doubtful whether it forms a suitable means of control for such purposes as motor vehicles, owing to its influence either on the gravity-fed compensating jet or upon the diffuser when such jets are used. For carburettors with mechanical compensation it would no doubt work admirably.

The sixth condition, namely, that it shall be simple to adjust in the first instance, and unlikely to get out of adjustment in use, is a very important one indeed, because in practice it is often very difficult to determine whether the metering is correct or not--too weak a mixture, either throughout or at any point in the range, shows itself at once by backfiring into the induction system, but it is not always so easy to make sure that the mixture is not too rich. Carburettors in which a gravity-fed compensating jet is used are always rather difficult to adjust accurately in the first place unless the engine is run under conditions wherein its fuel consumption, &c., can readily be determined at all loads and speeds, as on the test bench. On the other hand, such carburettors, once adjusted, will remain in adjustment for all time.

Carburettors with moving parts controlling the compensation are, on the whole, much easier to adjust in the first instance, but are liable, owing to wear, leakage, &c., to lose their adjustment. Finally, there are, on the market, several carburettors which, though they do not under any circumstances give correct metering, are none the less so easy to adjust to give a fair average proportionality, that in unskilled hands they are often found to yield results as good as or better than those of more scientific design.

As stated previously, there is, so far as the author is aware, no single carburettor which conforms to all the conditions he has laid down; but despite this, a good modern carburettor when properly adjusted is a remarkably efficient piece of apparatus. Though it is often much abused, the fault more often lies with the distribution system than with the carburettor itself.

It must be remembered always that the carburettor and the distribution system are closely interdependent. When the distribution system, either by its large exposed surface, the low velocity maintained in it, or its low temperature, is such as to encourage precipitation, then it is desirable to use a carburettor which will give very thorough pulverization, even at the expense of some loss by wire-drawing, and which will supply a large excess of fuel when the throttle is opened suddenly. On the other hand, with a different distribution system quite other characteristics may be desirable. It is quite wrong to suppose that any standard carburettor may be tacked on to any existing distribution system without regard to the particular characteristics of either.

Down-draught Carburettors.—The down-draught carburettor, which is a comparatively recent development, has important advantages which may be summarized as follows:—

(1) By virtue of its position above the manifold and above the hot spot, a slower gas velocity and thus a larger choke-tube may be used, considerably reducing the wire-drawing effect of the carburettor and giving an improvement in volumetric efficiency.

(2) The tendency for the heavier fractions to lag behind the lighter constituents of the petrol is reduced.

(3) A greater proportion of the petrol may be vaporized outside the cylinder and at a smaller cost in general induction air heating.

(4) Because of (3), a more homogeneous fuel/air mixture is prepared by the carburettor and hot spot and, in consequence, a more uniform mixture is distributed to the various cylinders.

(5) The carburettor is in a more accessible position for adjustment and cleaning.

These advantages depend for their realization on the combined use of a down-draught carburettor fitted vertically above a hot spot located in a central junction chamber from which there is easy access to the induction manifold running along the engine and feeding the various cylinders. The hot spot usually consists of a thin sheet of material of high thermal conductivity, such as copper, siamesed between the exhaust and inlet manifolds, which are given corresponding apertures for the purpose. The heavier drops of fuel which are not vaporized in the air stream fall on to the plate and are at once evaporated and carried by the air stream along the induction manifold. Contact between the unvaporized fuel droplets and the hot spot is thus assured, and complete vaporization takes place with a very small application of heat to the charge.

The whole ensemble of carburettor, hot spot and inlet manifold should be designed as one unit, so that the benefits outlined above may be fully realized.

It should be emphasized that the greater part of the life of the vehicle engine is occupied in running with the throttle about ¼ open and the engine developing something like 20 to 30 per cent of its full power.

Reference to the curves of road performance in Chapter X shows that the power output of a 2-litre engine propelling a 31-cwt. motor-car at a steady 35 M.P.H. on top gear is of the order of 10 horse-power. The engine speed at this gear ratio is 2100 R.P.M., and the full throttle B.H.P. 40. The engine load factor is therefore 25 per cent. The throttled fuel economy is therefore of great importance in determining the performance of a vehicle in ton-miles per gallon, and recently a good deal of attention has been given by engine and carburettor manufacturers to this aspect of vehicle operation. Experience has shown that the provision of an over-rich mixture for starting and idling and of a small reservoir of fuel for acceleration, while contributing much to the liveliness of the car, has little effect on the overall fuel economy, but that a weakening of the mixture on part throttle has a profound influence.

Owing to the relatively high manifold depression when throttled, early vaporization of the fuel ensures good mixing, with the result that the range of mixture strength may be extended further to the weak side before the occurrence of hesitant firing or actual missing. To enable full use to be made of mixture control at part throttle by special devices, such as bleed holes brought into operation below a certain manifold depression, further assistance may be obtained from a sparking-plug gap which is more than double the traditional value of 0·015 in. to 0·020 in. and used with high-voltage coil and special contact-breaker and circuit.

Of recent years refinements of this kind and test-bench calibration and adjustment of carburettor jet sizes have given opportunity for improving the ton-mileage per gallon of fuel by some 20 to 30 per cent. Taking the example quoted above, a ton-mileage of

40 per gallon would constitute a good performance. Such a road consumption would be given by an engine of C.R. 6 : 1 developing on a fuel of 70 octane number fuel consumptions of the order of 0·51, 0·56 and 0·78 pt./B.H.P. hour at $\frac{3}{4}$, $\frac{1}{2}$ and $\frac{1}{4}$ full-load respectively. These values assume a manifold system giving good distribution between cylinders.

Ignition.—By a process of elimination, ignition systems for high-speed internal-combustion engines have been reduced to two main types:

(1) The high-tension magneto.

(2) The high-tension coil and battery system.

The former was in almost universal use until a few years ago; the latter is a reversion to an earlier type, and has come into prominence again because nearly all vehicles and aircraft are now equipped with electric-lighting sets, so that an ample supply of low-tension current is always available.

Intensity of Spark.—Although when working on full throttle with a " correct " or slightly over-rich mixture the intensity of the spark is of very little importance, yet on reduced loads or weak mixtures the intensity plays a very prominent part. It has been found experimentally that when the nature, conditions, or consistency of the fuel/air mixture are such as to yield very rapid burning, then a very feeble spark will suffice. So far as maximum power output alone is concerned, the intensity of the spark does not play any serious part; it has, however, a considerable influence upon the maximum efficiency attainable, since it governs, to some extent, both the range of burning on the weak side and the rate of burning when throttled.

In the ordinary high-tension magneto, the intensity of the spark is at a maximum only at the point of maximum flux, and is therefore reduced when the time of ignition is either advanced or retarded beyond this point. In the coil and battery system, on the other hand, the intensity remains the same irrespective of the time of ignition. This is a substantial argument in favour of the latter system.

In the past, the coil and battery system was superseded by the high-tension magneto on the score of reliability for two reasons:

(1) Because there was no charging dynamo available, with the result that the battery was not kept fully charged and was frequently allowed to run down, with consequent complete failure of the ignition system.

(2) The early low-tension contact-breakers were generally ill designed and badly made, and gave continual trouble.

When the high-tension magneto first appeared on the market it had two substantial advantages over its rival system: namely, a constant supply of low-tension current, and a really well-designed and well-made low-tension contact-breaker.

These advantages are not so prominent to-day, for nearly every high-speed engine is equipped with a charging dynamo for lighting and starting.

It is not proposed to discuss the details of either system, for these are now well known and can be found in any text-book on the subject.

Sparking Plugs.—Probably no part of an internal-combustion engine is more complained of and abused than the sparking plug, though the complaints levelled against it are often unjust. It is generally complained either that sparking plugs oil up and so become inoperative, or that they give rise to pre-ignition; but the fault quite as often lies in the choice of an unsuitable plug for the engine or in defective design, with the result that too much oil passes into the combustion chamber.

In any type of sparking plug there is a limited range of temperature between which the points will be sufficiently hot to burn off any oil which may be deposited upon them, and sufficiently cool to avoid pre-ignition.

In those engines in which, owing to defective piston design, the quantity of oil passing the piston is excessive, it is necessary, in order to burn off the oil, to employ a type of plug with thin points which will keep hot when running on reduced loads. In this connection it should be remembered that, in a throttle-controlled engine, the actual temperature of the working fluid is nearly the same at any throttle opening, and that it is the total quantity of heat and not the temperature which varies. If the plug points are thin and their facilities for getting rid of the heat are poor, they will attain a temperature corresponding to the mean temperature of the cycle irrespective of the quantity of heat liberated. If, on the other hand, they are provided with good facilities for getting rid of the heat imparted to them, then their temperature will be governed rather by the quantity of heat, but will at all times be somewhat lower. In other words, in plugs with long thin points the points will always be hot but will remain at much the same temperature at any load; while in plugs with short, thick points and good facilities for getting rid of their heat, the points will keep cooler at all loads, but their temperature will vary over a wider range as the load is varied. From the point of view of maintaining an equable temperature at all loads, comparatively long, thin points are preferable, and so long as the compression is low and there is no detonation to increase the temperature and rate of heat flow they will not give rise to pre-ignition. But when the compression ratio is high and the proportion of diluent is therefore small, pre-ignition will occur the more readily; moreover, under these circumstances detonation is the more liable to occur, and this, as has been shown previously, will give rise to overheating of the plug points. For such engines, therefore, it is necessary to use a type of plug whose points will keep as cool as possible. When, by careful design, the flow of lubricating oil· to the combustion chamber is reduced to the lowest limit, it becomes possible to use a " cool " plug without trouble from oiling up, and it is then preferable to do so, for the thick points naturally last longer, and there is less risk of overheating from momentary detonation or other causes.

For low-compression engines or for engines whose duty is comparatively light a " hot " plug may be used with advantage, more especially if they have any tendency to pass an excess of oil; while for high-compression engines a " cool " plug must be used, and the tendency to oil up must be overcome by adequate piston design. No single plug can at present be made to suit a high-compression engine which passes excess oil into the combustion chamber, but the remedy lies in the design of the engine, and it is not fair to abuse the sparking plug because it is being called upon to meet conditions outside its legitimate range.

Sparking Plugs for High Duty Engines.—Special types of sparking plug have recently been developed to resist corrosion and the high intensity of heat flow of the high duty engine. The following features enable them to meet modern requirements:—

(1) They are of small diameter, usually 14 mm. The plug boss is therefore small and easily cooled by adequate water passages.

(2) They have a robust central electrode, which occupies a very large proportion of the internal volume of the plug. The material of the electrode must be predominantly a corrosion-resisting alloy and it usually contains a high percentage of nickel.

(3) The nose end of the central electrode is almost flush with the cylinder wall, recessed slightly to allow room for the earth wires without exposing it to the rigours of blasts of hot gas in the chamber.

There is therefore practically no space into and out of which gases may surge and overheat the plug terminals.

(4) The earth terminals consist of fine platinum alloy wire, consisting of a high proportion of platinum with 10 or 20 per cent of iridium. This alloy has a very high melting-point and is highly resistant to corrosion and erosion. The low heat capacity of the fine wire enables it to follow the cycle temperature. Its high temperature during combustion then reduces the tendency to " oil up ", while the fact that cooling occurs during the idle strokes prevents the wire from being a cause of pre-ignition.

(5) A new insulating material is used, consisting of a sintered aluminium compound which is a good conductor of heat and is not attacked by lead at high temperatures.

CHAPTER VI

MECHANICAL DESIGN

In the design of an internal-combustion engine, as in all creative work of this nature, the æsthetic side must not be overlooked. In the first place, beauty of form and of proportions is in itself an admirable guide to mechanical correctness; for mankind has come to regard as beautiful that which is mechanically correct, whether it be in nature, in architecture, or in engineering.

In general, beauty and efficiency in the widest sense of the term are synonymous, and the appeal of any design to the æsthetic sense is often as reliable a guide as is a mathematical analysis of its mechanical features. Again, the æsthetic side makes a powerful though an unconscious appeal to the user, whose artistic sense, mute and inarticulate though it may be, will always be roused.

The designer's first aim should be to ensure that the products of his work will receive the care and even affection which he hopes will be bestowed upon them by their users. To this end he should make an appeal to them through their artistic sense rather than to fads or fashion, for the former is innate in all mankind, while the latter may vary widely.

There is a prevalent but quite erroneous belief that the reliability and even the efficiency of an engine are, to a large extent, a function of the actual number of parts it contains. Speaking generally, there can be no greater fallacy. While it is obvious that the number of parts should be kept down to the minimum compatible with efficiency and mechanical correctness, this can very easily be overdone. Fewness of parts too often denotes excess of compromise. All design must necessarily be based on compromise, and it is upon the soundness of judgement by which the compromise is arrived at that the success of an engine ultimately depends. In an internal-combustion engine many of the parts are subjected to complicated stresses, both heat stresses and pressure stresses; and when, by multiplying the number of parts, such stresses can be reduced or dealt with separately, this should be done unhesitatingly. No member should be subjected to compound stresses if by the provision of additional members these can be split up; for example, when a member is subjected to combined torsion and bending it is preferable, where possible, to replace it by two separate members, one designed to deal with the bending alone and free from torsion, and another subject to torsion only and free from bending. To do this may involve the introduction of perhaps six or eight times as many parts in this particular piece of mechanism, yet the safety and reliability will be many times greater. Again, in the design of an engine the problem often arises of carrying a shaft in two bearings whose perfect alignment with each other cannot be ensured in machining or is liable to disturbance in use owing to distortion, &c. In such cases the choice lies between fitting a fairly flexible shaft which will accommodate itself by flexure to slight errors in alignment, or the provision of double universal joints between the two bearings. In the former case the shaft is liable to fail ultimately from fatigue through constant flexure, however slight, while the bearings are liable to give trouble, and in any case the friction will be much greater. In the latter case safety and the minimum of friction are ensured, but at the cost of several extra parts. A choice of this nature confronts every designer almost daily, and he has to decide whether he will risk the simple expedient or resort to the more complicated one. He is generally too liable to adopt the former course on the ground of manufacturing cost, but in such case he has no right to boast that he is using fewer parts—he is doing so only because he cannot afford to use more, or because he has neither the knowledge nor the experience to appreciate the risk he is incurring. Again, in the larger sizes of engine it is often desirable to duplicate the exhaust valves, even though this may involve the duplication of the whole valve gear also. By doing so, smaller valves can be employed, and both the temperature of the valve heads and the stresses in the valve gear will be reduced, the valves will remain in good condition for a far longer time, while the margin of safety in the valve gear will be greatly increased. The net result of duplicating the exhaust valves will be that the engine will retain its efficiency for a much longer period, and will at all times be more reliable. It is better far to use 500 parts if need be, to comply with the laws of sound mechanics, than to defy them with a single part. The craving to make one single member perform several distinct functions is often very difficult to resist, but it should be firmly controlled.

It is possible to produce an internal-combustion engine composed of only seven parts (exclusive of studs and nuts). If fewness of parts were a criterion, such an engine should sweep the board. In practice this type of engine has earned almost universal condemnation because of its unreliability and the ingenuity with which it devises different ways of going wrong. At the other end of the scale the aero-engine which, during the war of 1914–18, showed itself capable of running for by far the longest period without overhaul, was the Rolls-Royce " Eagle ", an engine which contained at least 50 per cent more parts than any other engine in the service.

That increased number of parts necessarily involves increased care and maintenance on the part of the user is a sheer fallacy. Fewness of parts saves manufacturing costs to some small extent, but it certainly confers no benefit whatever upon the user. Even to the manufacturer it does not always effect a saving, for the amount of fitting work required is often inversely proportional to the number of parts, and fitting is nowadays the most costly of all classes of work. Designers will do well to realize that " simplicity " as ordinarily understood is by no means always a virtue ; in nine cases out of ten it is a positive vice.

The foregoing remarks must not be read as implying a disregard of manufacturing cost. On the contrary, manufacturing cost is generally much the most important consideration once the needs of all-round efficiency have been catered for, and by efficiency in this sense is meant not merely thermal efficiency but reliability and durability.

In days gone by, material and skilled fitting were both comparatively cheap, while tooling was a costly item. To-day, however, owing to the vast improvements in machine-tool design, tooling and grinding have become relatively cheap, while the cost of material has risen enormously and skilled fitting has become almost unobtainable. The designer therefore must accommodate himself to these altered conditions and economize material and hand-fitting wherever possible. In this connection he will often find that it is an actual economy to employ a greater number of parts if the total weight of material used is no greater and some hand-fitting is saved thereby. With well-thought-out design and accurate machine work it should be possible to eliminate hand-fitting almost entirely. Much of the hand-fitting which is done to-day is unnecessary and even undesirable; the scraping-in of bearings, for example, is a custom which dates from the time when accurate machine work could not be relied upon and when designers did not fully realize the value of alignment. Now that crankshafts can be finished by grinding to within extremely close limits, and all their bearing housings can be machined at one operation, scraping is no longer necessary; indeed, no amount of hand-scraping will give so accurate or uniform a bearing as that afforded by the machining. Again, from an economical as well as from a mechanical point of view it is essential that all shaft-bearing housings shall either be machined at one operation or their alignment be ensured by spigoting. If, for any reason, neither of these is possible, then it is not only better but often even cheaper to provide universal joints and so be independent of alignment, rather than to rely upon a doubtful alignment secured by costly hand-fitting. Not only is hand-fitting expensive and unreliable, but it is also the most effective barrier against interchangeability. The old belief that a " hand-made " piece of mechanism is preferable to a " machine-made " died hard; to-day such a belief is an anachronism.

Design and Material.—It is commonly supposed that, to be successful, a high-speed internal-combustion engine must of necessity be made from very carefully selected and highly specialized materials. While, of course, it is obvious that the higher the quality of the material the better, yet, with appropriate design, the ordinary materials of commerce will be found to give perfectly satisfactory results, and are, in fact, much more widely used than is generally supposed. It is only when the weight is very closely limited, as in the case of an aero-engine, or when an exceptionally high output is desired, that fancy materials become necessary.

The increase in specific power output in all types of internal-combustion engines, however, has created a need for materials able to withstand more and more severe working conditions. Great advances have been made in the production and cheap supply of alloy and special steels and of the lighter non-ferrous alloys, to such an extent that the whole aspect of design has changed. The fancy material of yesterday has become the essential of to-day. High-speed engines of moderate output, as judged by present-day standards, now incorporate a higher proportion of high tensile and heat-resisting steels and light alloys having special qualities, such as low thermal expansion and high load bearing capacities for the working parts. Many of these new materials were unknown a few years ago. In addition, surface hardening processes have been developed which have gone far to meet the need for surfaces which will stand up to high duty without undue wear.

The ordinary multi-cylinder high-speed engine necessarily involves the use of some very complicated castings, and the choice of material for these is, in practice, governed almost solely by foundry considerations. The material of the cylinder block and crankcase, for example, must be such that it will flow freely in the mould and yield a homogeneous casting, a consideration which generally dominates all others.

In general, surface hardness is of more importance than tensile strength; for the necessity for extreme rigidity, which is the first principle in high-speed engine design, compels the use of such heavy scantlings in any case that tensile strength plays a very secondary part. From the point of view of rigidity all steels are substantially equal, for the modulus of elasticity is practically the same for all. The increased stress range sustained by the working parts in the modern engine, however, requires the use of steels of high tensile strength to give an adequate margin for fatigue.

Crankcase.- For the crankcase either cast-iron or aluminium alloy may be used, according as weight is an important factor or not. The chief consideration in the design of a crankcase for a multi-cylinder engine is to provide sufficient depth to resist bending moment due to the opposed couples in a four- or six-cylinder engine. Severe hogging and sagging forces are set up in a vertical plane, and the centrifugal forces due to the outer and inner groups of cylinders set up bending stresses in alternate directions in a horizontal plane.

The bending moments are not so severe as to affect the structural strength of the crankcase, but they are often severe enough to cause slight deflections which give rise to noise and vibration and a kind of indeterminate and elusive roughness in the running of the engine. In order to prevent this it is necessary to employ a design of crankcase so strong to resist both vertical and horizontal bending and torsion that from a strength point of view it has an enormous factor of safety. Since rigidity is a function of design rather than of material, it follows that, within limits, any material which will cast well and possess reasonable physical qualities will serve the purpose.

Experience has shown that even when cost of manufacture is of importance, it is a sound principle in the design of the crankcase not to place much reliance on the stiffness lent by the cylinder block and lower half crankcase as an aid to rigidity when these are separate entities. At first sight it might seem that much stiffness might be borrowed from these members if they were securely bolted by many bolts to the top and bottom of the crankcase through stout flanges. In practice, however, it is found that the better and cheaper solution is to make the crankcase self-supporting in structural rigidity. It is then made as deep and as robust as possible, and to resist bending stresses in a horizontal plane it is either given a bulging shape, when seen in plan view, or is provided with a horizontal stiffening web running the length of the engine in the plane of the crankshaft. Another method of giving resistance to horizontal deflections is to make the crankcase double-walled with a kind of blister on each side.

In these days of improved technique in the foundry and with greater need for smoothness of operation, the practice is widely followed of extending the crankcase up to the cylinder head to include with it the cylinder block in one casting. In general, this is a rather heavier construction than a separate cylinder block design, because the casting is more complicated and requires more generous allowances at inaccessible points. An example of an integral cylinder-crankcase construction is shown in fig. 74. In this case a light-alloy sump of very light construction is all that is necessary to enclose the lower part of the engine and hold the lubricating oil. By using wet liners, narrow journal bearings and close cylinder centreing, a comparatively light compact engine for commercial vehicles is achieved, weighing no more than 12 lb. per horse-power at maximum power output, and 15 lb. per horse-power at a rated full-load of 80 per cent of the maximum. The crankcase and cylinder heads are in cast-iron, while the various covers, such as the valve gear and timing covers, are in aluminium alloy.

Another consideration which must always be borne in mind is the question of noise due to the vibration of flat panels, &c. As far as possible all flat surfaces should be avoided, or, if this is not convenient, they should be broken up by internal ribbing. There is some evidence to show that for similar designs and scantlings cast iron is more resonant than aluminium. In the first engines which the author designed for tanks in the war of 1914-18, cast iron was used for the crankcases, but later on, when light high-speed tanks were called for, aluminium was substituted. These were made to exactly the same design as the previous cast-iron cases, and were, in fact, interchangeable with them in every respect. It was noticed after this change that the engines were appreciably

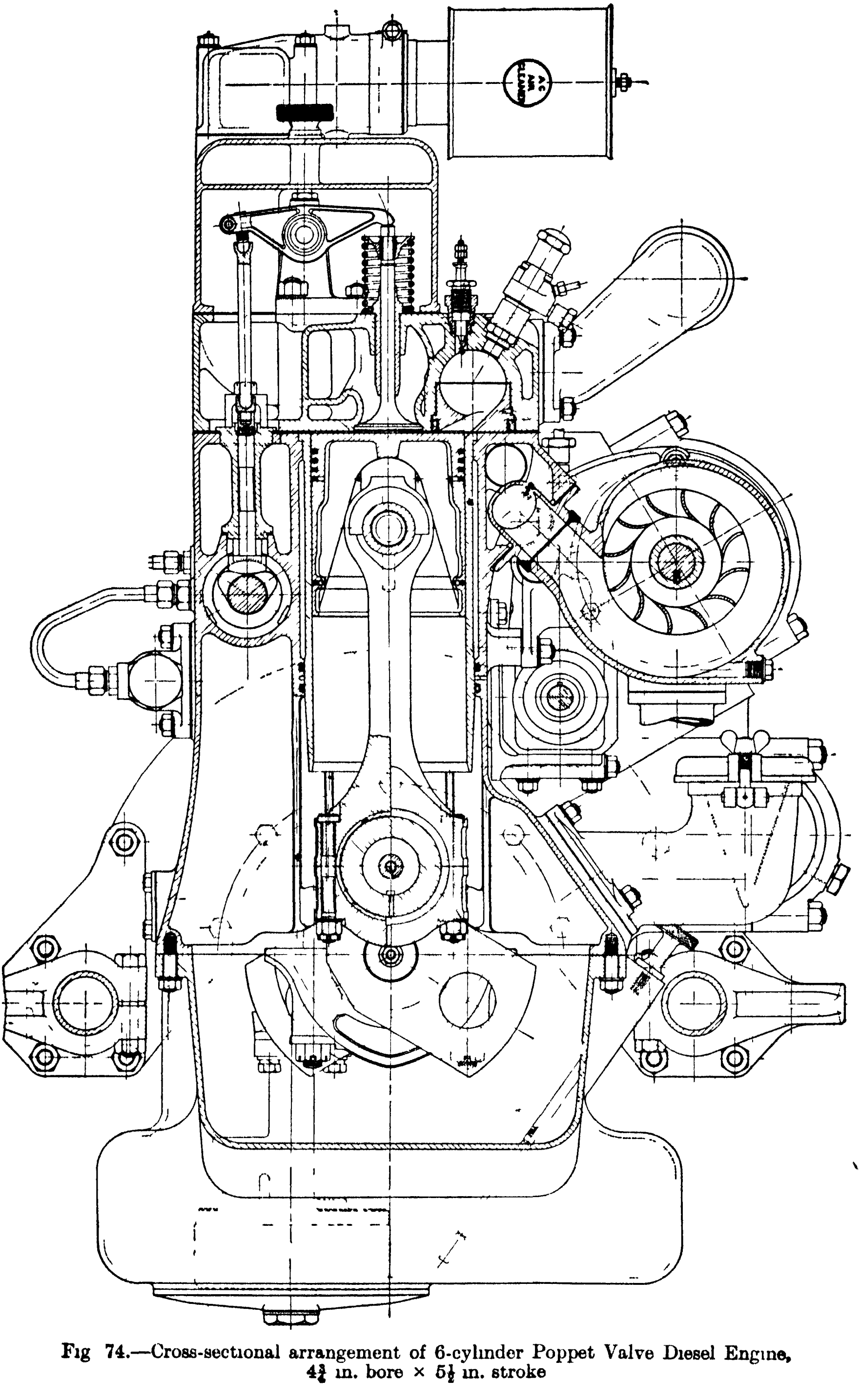


Fig 74.—Cross-sectional arrangement of 6-cylinder Poppet Valve Diesel Engine, 4¾ in. bore × 5½ in. stroke

quieter in operation, a fact which was later confirmed by the reports of officers and drivers in charge.

There is another important consideration in favour of the use of aluminium for crankcases; namely, its better heat conductivity. This is a very valuable feature, because it more readily leads away and dissipates the heat from the crankshaft bearings, and so tends to reduce wear and the risk of bearing failures.

The principal objections to the use of aluminium are its greater cost and the difficulty in securing studs in it. The former is not so serious an objection as would appear at first sight, because:—

(*a*) For complicated castings aluminium alloys are, on the whole, more reliable from a foundry point of view than cast iron; hence the proportion of wasters, often only discovered after several machining operations, is reduced.

(*b*) The cost of machining aluminium is considerably less than that of cast iron; and since in a crankcase there are usually a very considerable number of machining operations, this often outweighs the higher initial cost of the material.

The difficulty in securing studs in aluminium is a real one, but it is usually possible so to design a crankcase as to eliminate or almost eliminate the use of studs. By extending the main crankshaft bearing bolts through to the top of the crankcase and utilizing them to hold down the cylinder block, two groups of studs can be eliminated and a very sound mechanical job results. Where studs must be used they should be screwed in from 2 to 3 diameters, and are then satisfactory, if properly fitted in the first place.

Cylinder Block.—Where a number of cylinders are cast together in one block, cast iron is almost invariably used, though recently aluminium blocks fitted with steel or cast-iron liners have been adopted for aircraft engines and for a few motor-car engines also. So far as the plain cast-iron block is concerned there is little to be said except that two considerations should be aimed at in the choice of a material; that it shall cast readily and be free from blow-holes or porous places, and that it shall be as hard as possible compatible with the first consideration. There appears to be no merit in the use of the close-grained iron so often called for in specifications, for an open grain may be just as good and is sometimes better. In this connection it may be mentioned that cylinders finished by grinding seldom wear so well as those which have been reamed; this appears to be due to the retention of small particles of the grinding material in the surface of the bore. These soon embed themselves in the piston, which then proceeds to lap the cylinder walls, more particularly when the piston is of aluminium alloy. The habit of grinding cylinder bores came into force at a time when it was found difficult to obtain a satisfactory finish by tooling alone; and it was soon found to be both a cheaper and a more satisfactory means of obtaining at once a reasonably accurate bore and a good finish, and so found wide favour in the eyes of manufacturers. With modern improvements in machining methods it is now possible to obtain excellent results by reaming and honing, and this process would appear to be preferable. With care and a suitable choice of grinding wheels, embedding of the grinding material in the cylinder bore can be avoided.

In recent years it has become the practice to make the whole cylinder-head detachable. This practice has many important advantages. In the first place, it simplifies the cylinder block casting very considerably and enables the cylinder to be bored straight through—a very substantial advantage. Secondly, it permits of the valves being placed closer together, and so, in side-valve engines, reduces the area of surface in the valve pockets and permits of the cylinder centres being reduced, both very valuable considerations.

The principal objection to detachable heads is that they necessitate the making of a gas- and water-joint over a large area and the breaking of this joint in order to get at the valves or clean out the combustion chambers. By the use of suitable copper asbestos gaskets, by providing plenty of holding-down studs suitably spaced, and an ample depth of head, these objections may be overcome. Where trouble with cylinder-head joints has arisen, it can generally be traced either to lack of rigidity in the head or to insufficient or badly placed holding-down studs. Given careful design and ample rigidity, especially in the region stressed by the explosion, the use of detachable cylinder-heads has much to recommend it. Where weight is of first importance and gas pressures are high, however, as is generally the case in aëro-engines, the integral or permanently-attached cylinder head must still be used, since it affords a greater measure of reliability.

At one time it became the custom to embody the induction manifold in the cylinder block casting. This certainly makes for neatness, and, by reducing the number of joints in the system, lessens the tendency to leakage; but there is nothing else to recommend the practice, which has three serious defects:

(1) The design of the induction system, and therefore the efficiency

of distribution, must be subordinated to foundry requirements. It is seldom, if ever, possible to design an efficient distribution system which can be embodied in the cylinder block casting.

(2) With modern petrol having a final boiling-point of about 350° F., jacketing of the induction system with warm water is of very little use; it merely serves to supply heat to the working fluid before its entry to the cylinder, without ensuring evaporation of the heavier fractions. In previous chapters it has been shown that the less heat supplied to the gases before their entry to the cylinder the better. To prevent precipitation some heat must be supplied, but to be of real use it must be supplied at a high temperature. It is far better, therefore, to supply locally a small quantity of high-temperature heat just at those points where, owing to changes in velocity or direction, precipitation of liquid fuel is most liable to occur, rather than to subject the whole system to a continuous supply of low-temperature heat, which serves merely to reduce the power output without evaporating any but the lighter fractions of the fuel.

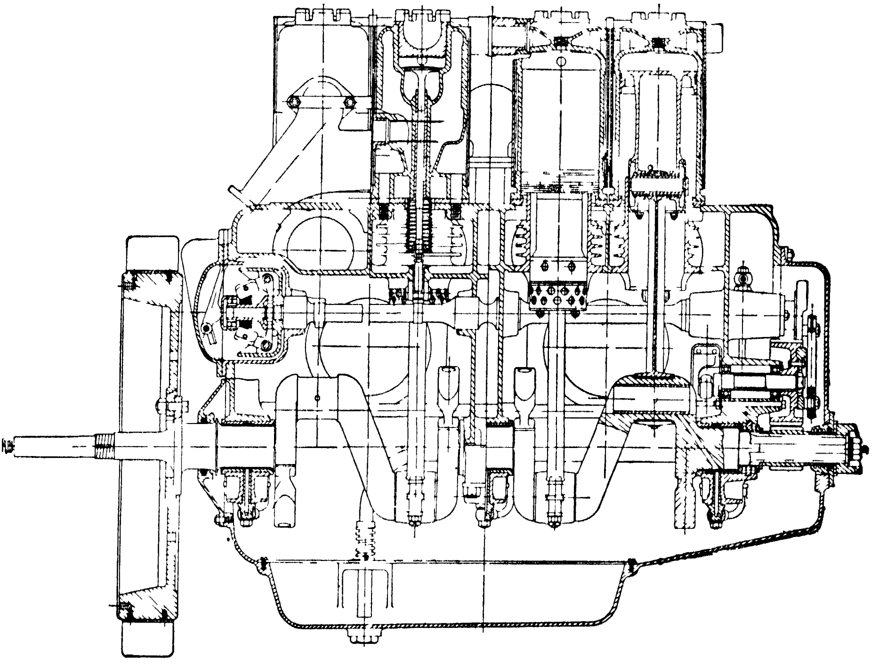


Fig 75 — Section of Four-cylinder Engine, showing Open-sided Cylinders

(3) When the induction system is cast in the cylinder block the whole of its internal surface is rough; this greatly increases the

tendency of the liquid particles of fuel to precipitate, and it increases also the frictional losses.

Apart from the orthodox multi-cylinder monobloc type, there are many other forms of cast-iron cylinder. Fig. 75 shows a form—becoming less common—embodying separate cylinders in which the sides of the water-jacket are cast open and subsequently closed with thin sheet-steel covers. This has the advantage that the casting is very simple, can be inspected internally, and the core sand thoroughly removed. Also, it permits of a number of cylinders being packed together more closely than would otherwise be possible with separate cylinders, thus both reducing the length of the crankshaft and increasing the rigidity of the crankcase, as shown in fig. 76.

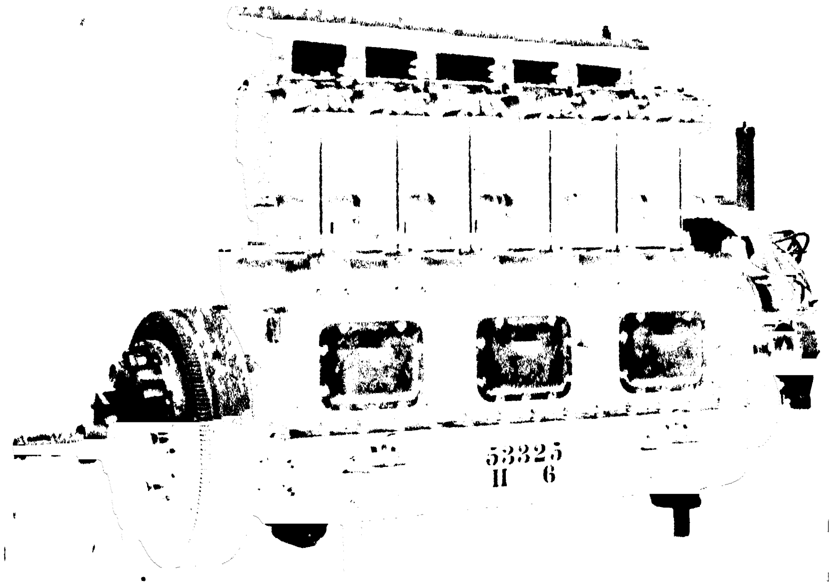


Fig. 76.—225 B.H.P. Tank Engine, showing close packing of Cylinders

Fig. 77 shows a design for large cylinders, in this instance of 100 horse-power, in which the head consists merely of a circular plug carrying the valves, which can be removed very readily and with the minimum of disturbance to pipe joints, &c.

There are a great many different forms of cylinder construction in use for aircraft engines. One form of cylinder block construction which the author favours for very light or high-duty engines consists in carrying the aluminium crankcase up to the cylinder head and inserting in it loose steel liners machined all over and provided with flanges at the top nipped between the top of the crankcase and the cylinder head. The lower end of each liner passes through a diaphragm in the crankcase which forms the bottom of the water-jacket, the water-joint being made by means of a rubber ring as in gas-engine practice. This system, actual forms of which are shown in figs. 78 and 79, is of course only an adaptation of the ordinary

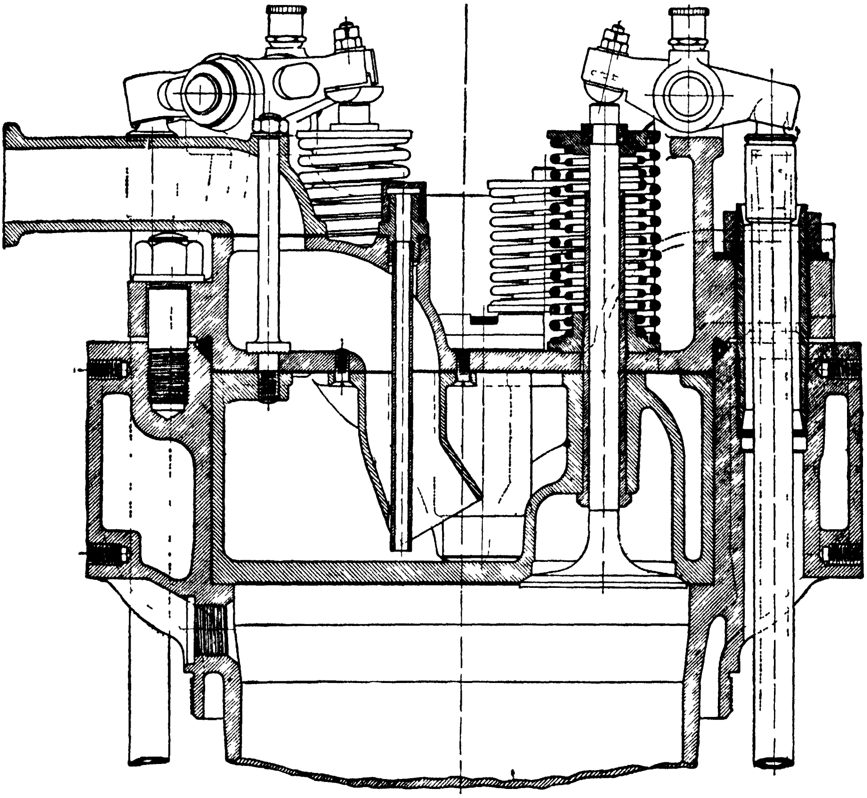


Fig 77.—Section of Cylinder and Head, 100 B.H P. Engine. 8¼-inch bore, [illegible]½-inch stroke. Speed, 1150 R P.M.

horizontal gas-engine construction, but it has, in the author's opinion, the following advantages:—

(1) It is at once both light and inexpensive.

(2) It renders the top half of the crankcase very rigid against bending or torsion.

(3) It is simple and easy to cast.

(4) The water connections can be secured to the main casting, and not to light sheet-metal jackets.

Yet another form of cylinder construction which is sometimes favoured consists in casting the whole cylinder block in aluminium alloy with cast-in valve seats, and either screwed or pressed-in steel liners. This construction is open to the objection that it is always difficult to ensure good thermal contact between the liner and aluminium walls, on account of the very great difference in the coefficient of expansion of the two materials.

A form of cylinder construction which once found great favour in aircraft engines is that in which both the cylinders themselves and the water-jackets are built up out of steel by welding. In such a construction it is customary to forge the cylinder and cylinder head as a plain thimble, to screw and weld the valve ports and valve guides into this, and finally to weld over all a light steel water-jacket pressed from thin sheet and made up in two halves. This form of construction is very light and sound, and thanks to the absence of castings, and therefore to the small proportion of scrap, it is not nearly so expensive as would appear at first sight. The chief objections to it are:

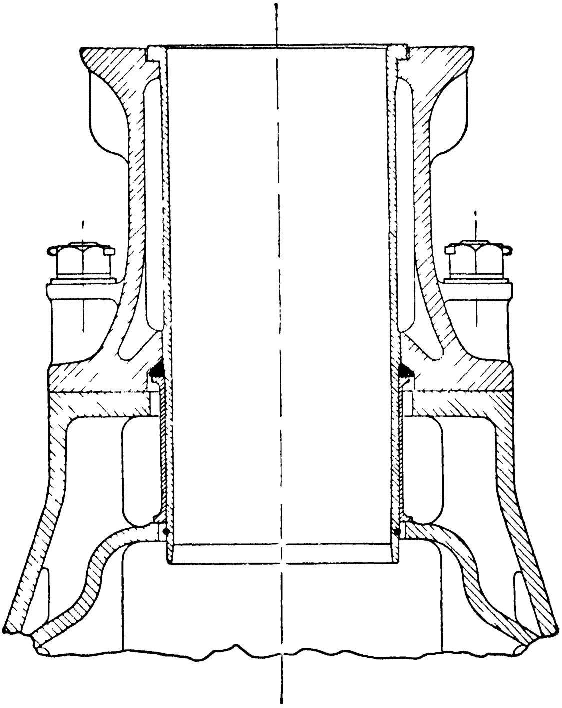


Fig. 78 —Section of Cylinder showing loose hardened Steel Liner sealed by Rubber Ring

(1) It is applicable only to separate cylinders, or at least it

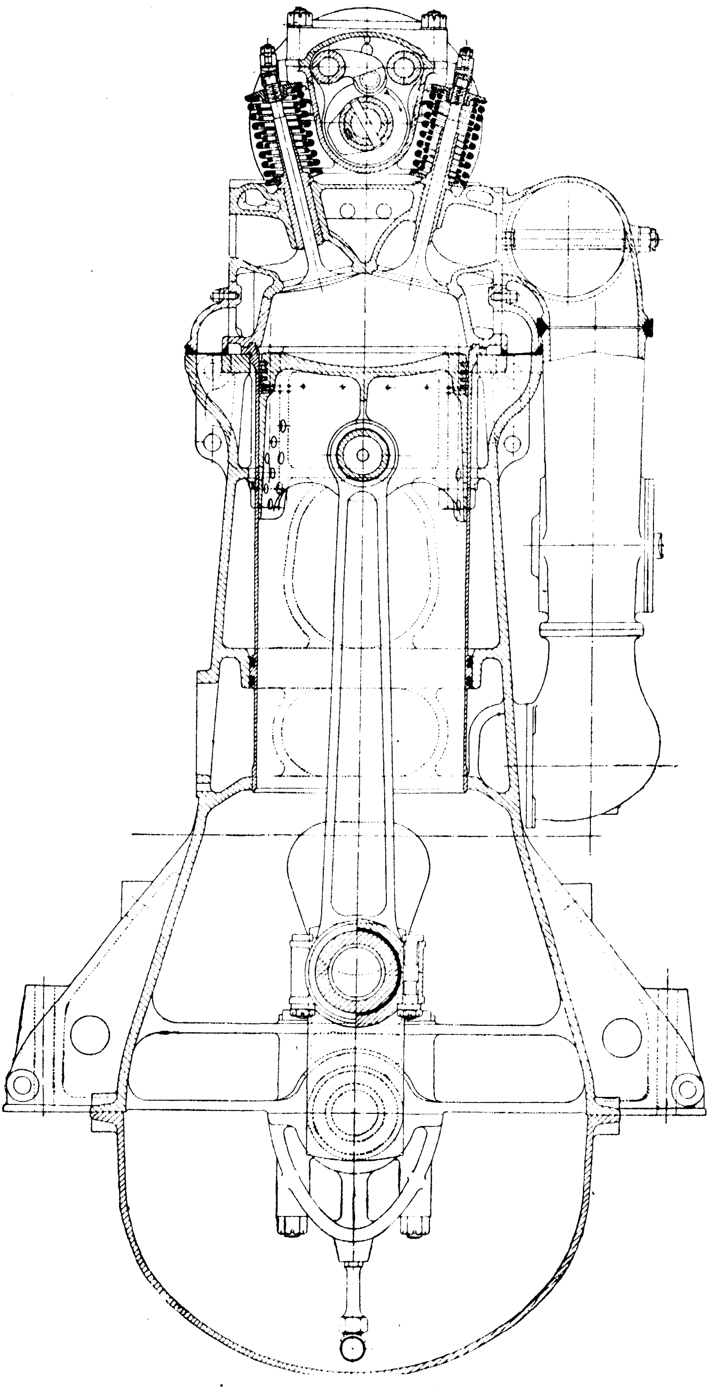


Fig. 79.—Section showing cylinder construction of large Experimental Aero-engine developing 130 B.H.P. per Cylinder. Bore 8 inch, stroke 11 inch, speed 1400 R.P.M.

becomes very complicated when the water-jacket embraces more than one cylinder.

(2) The water connections are attached only to light sheet-metal jackets, which are liable to crack from vibration.

(3) Freedom of design is rather limited by constructional difficulties.

Crankshafts.—There is always a considerable difference of opinion as to the most suitable material for crankshafts. Except in the case of aircraft engines, tensile strength and yield point are factors which need hardly be considered, for by the time the requirements of torsional rigidity, freedom from whipping, stiffness and rigidity of crankpins and journals, and the provision of adequate bearing surface in a limited length have all been met, it will be found that the scantlings of the shaft are such as to put all risk of failure from any source but fatigue quite out of the range of probability. The two essential requirements are resistance to fatigue and surface hardness. In the author's opinion it is almost a waste of time to calculate out the stresses in a crankshaft, for, if the design is adequate from the points of view enumerated above, it will invariably be found that the calculable stress in the material is quite absurdly low. It must be remembered that the modulus of elasticity for all steels is substantially the same, and since it is rigidity which controls the design of a crankshaft, all steels are nearly on a par in this respect. In the author's opinion a straight carbon steel containing about 0·45 per cent carbon fulfils all the requisite conditions quite satisfactorily for most commercial purposes except for aircraft engines. These latter are excepted because—

(1) Having no flywheel, aircraft engines are less susceptible to torsional oscillations in the crankshaft, and the latter can therefore be made considerably lighter.

(2) In order to save weight in the shaft and in the design generally, a bearing is always fitted between each crankthrow, hence the tendency to whip is greatly reduced.

(3) Owing to their relatively short working life, the area of bearing surface is generally cut down in order to reduce weight.

Apart from the question of cost of material, the advantages of using a straight carbon steel for ordinary engines are:

(1) That every engineering establishment is familiar with the methods of forging, heat treating, and machining it.

(2) That it is very uniform in quality, and much less sensitive to errors in heat treatment.

In other words, it is decidedly more dependable than high tensile alloy steels, all of which require careful heat treatment, and if wrongly treated, are liable to be dangerous.

Fracture of a carbon steel crankshaft of reasonable design is a very rare occurrence indeed, and is generally due either to want of adequate fillets at the crankpins and journals, or to periodic torsional vibrations; either of these causes will result in eventual fracture from fatigue, no matter what the material may be. The former may be avoided by providing an ample radius, and the latter by avoiding periodic vibration, either by fitting a vibration damper or by altering the scantling of the shaft so as to raise or lower the periodic speed out of the normal running range. In the case of four-cylinder engines with reasonably light reciprocating parts, it is generally quite easy so to design the crankshaft that there shall be no severe torsional vibration at any speed of which the engine is capable. In the case of six-cylinder engines, however, it is by no means so easy to do this, and it then becomes desirable to employ vibration dampers or other means to check torsional vibration.

The most important consideration to-day is to reduce the rate of wear. This, as shown previously when dealing with lubrication, is a function of the surface hardness, both of the shaft itself and of the bearing material in which it runs. The harder the shaft and the softer the bearing material the better, provided that the latter will not crush. Wear appears to be due almost entirely to particles of grit embedding in the soft bearing material and so lapping the shaft. The rate at which they will lap away the shaft depends upon (1) the surface hardness; (2) the load, which governs the thickness of the oil film and therefore the distance which the particles have to bridge before coming into contact with the shaft.

In general, the load on the crankshaft bearings is so severe that it is not always possible to use a very soft bearing material, but it should be borne in mind that, other things being equal, the softer the bearing material the less will be the wear.

Balance Weights.—In four- or six-cylinder engines, the centre crankshaft journal is subjected to very severe loading on account of the cumulative centrifugal and inertia pressure from the two cranks on either side of it, since these are always in the same plane; for this reason, either much greater surface must be given to this bearing, or the crankshaft must be fitted with balance weights to counteract the centrifugal pressure from the two centre cranks. Unfortunately, if the two centre cranks are fitted with balance weights, so also must the others. The provision of balance weights, however, while it relieves the load on the journal bearings, and more particularly on the centre bearing, and also reduces the tendency to vibration of the crankcase, is very objectionable because, by adding masses with considerable inertia to the various throws of the crank, it increases greatly the tendency to torsional vibration of the crankshaft, and at the same time tends to lower its critical speed. This may be, and often is, a serious objection to the use of balance weights, for torsional vibration of the crankshaft is much more serious in its effects and more difficult to deal with than that of the crankcase. The use of balance weights, therefore, is by no means always to be recommended; they may be of advantage in reducing wear, or they may set up severe and dangerous periodic vibration. Where, owing e.g. to restrictions upon length, the area of bearing surface of the centre bearing is limited, balance-weights are sometimes necessary, and their ill effects can then be counteracted by the well-known Lanchester torsional vibration damper.

There can be little doubt but that the ideal crankshaft should have hardened journals and pins. It is very difficult satisfactorily to case-harden a one-piece multiple-throw crankshaft on account of its tendency to buckle when quenched, except in the case of very short cranks such as are used in engines with two opposed cylinders. Modern methods of surface hardening, such as nitriding, are, however, less prone to cause distortion.

It is, however, in the author's opinion quite open to question whether the advantages of hardened bearing surfaces are not such as to justify the use of built-up cranks, with the webs shrunk on to the journals and the crankpins pressed or clamped into place. Fig. 80 shows an actual example of a crankshaft constructed in this manner for an engine of 125 B.H.P., running at a normal speed of 4000-4500 R.P.M., which has been found to give very satisfactory results. Fig. 81 shows an alternative design with the crankpins clamped into place, and roller bearings used throughout. This design is intended for an engine of more normal speed and performance. The bearings on such a shaft should prove well-nigh everlasting, but in the event of wear or any accident such as the fracture of a roller, the whole crankpin, complete with its roller bearing, could be replaced quite easily. Apart from their virtual immunity from wear, ball or roller bearings have the advantage that they are not dependent upon continuous lubrication to maintain the oil film and are therefore more reliable; also although the friction loss with plain

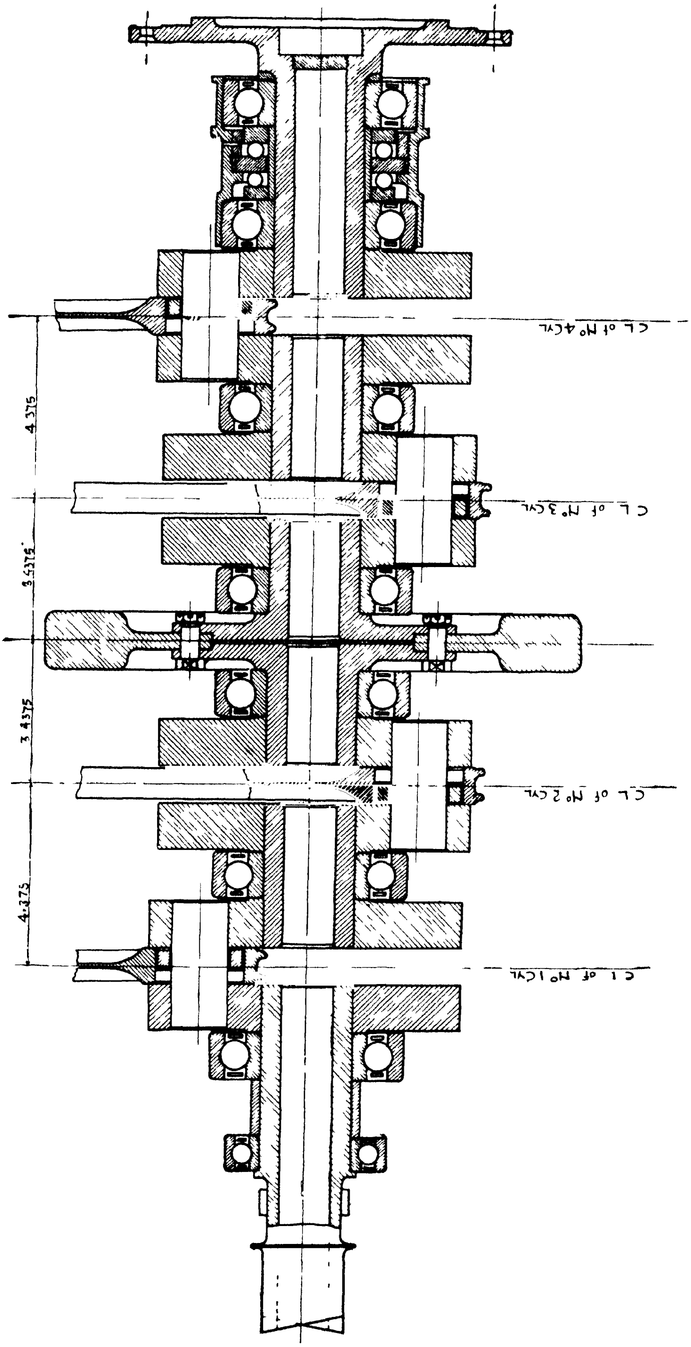


Fig 80.—Crankshaft Assembly as used in 3 litre 1922 Vauxhall Racing Engines

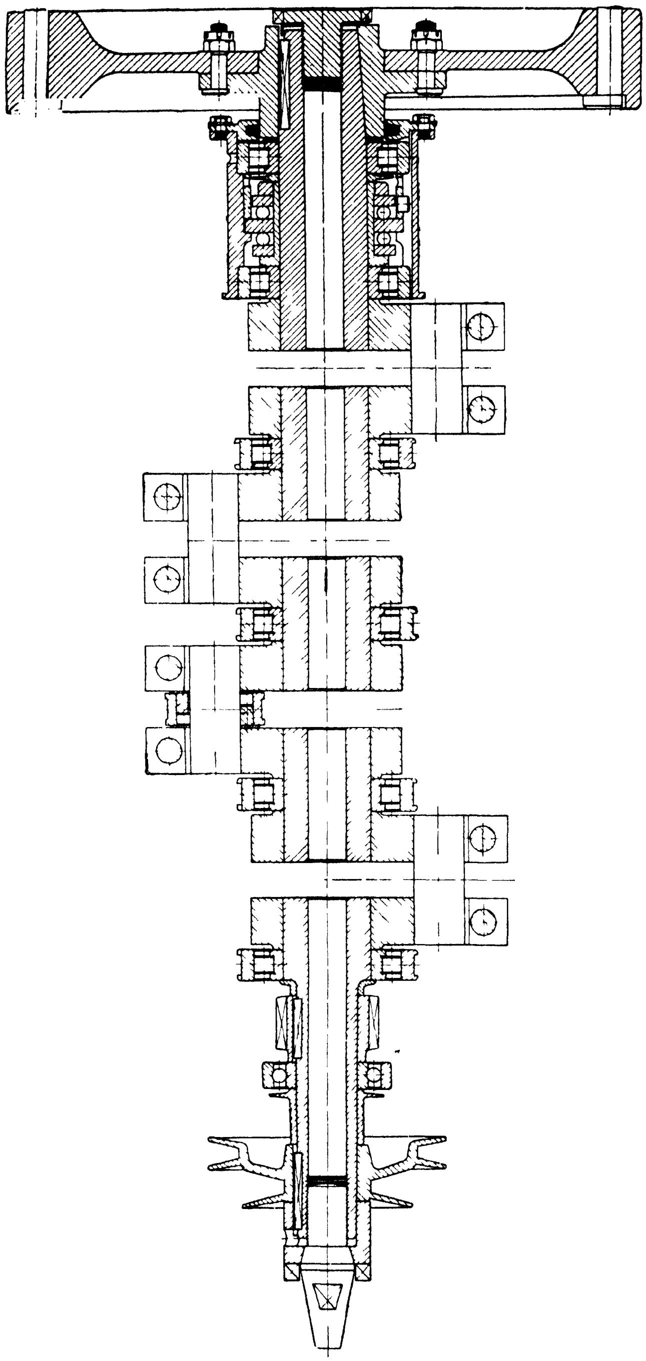


Fig 81.- Roller bearing built up Crankshaft for heavy Commercial Vehicle Engine

bearings amounts, in any average design, to less than 2 per cent of the maximum power of the engine at high speeds, and at normal working temperatures, this loss is almost independent of load, and is liable actually to increase as the speed is reduced. While, therefore, the total frictional losses of the plain bearings on a crankshaft are relatively small at high speeds and full load, at reduced speeds and loads they begin to play an important part. In the case of motor vehicle engines, which, for the bulk of their existence, operate under very light loads, and whose average load factor is only from 20 to 30 per cent. the use of a ball- or roller-bearing crankshaft would result in a very appreciable gain in fuel economy, probably about 6 to 8 per cent.

Wear of Crankshafts.—In general terms, it may be said that while the greatest wear in crankshafts generally occurs in the main journal bearings, actual failure of the bearing material is more frequent on the crankpins, even when both journal and crankpin bearings are subjected to the same load factor. The causes of this state of affairs are not far to seek. The greater liability to failure of the crankpin or connecting-rod big-end bearings as compared with the main journal bearings when both are subjected to the same mean loading per square inch is due to:

(1) The smaller heat capacity which the bearing has to draw upon.

(2) Inadequate support of the bearing in the connecting-rod, the big-end eye of which is generally lacking in rigidity.

In the event, therefore, of a temporary stoppage or slowing down of the oil circulation, the crankpin bearing will, in a given time, attain a much higher temperature than the main journal bearings, and, if the stoppage be prolonged, it will reach the critical temperature at which breakdown occurs (viz. about 360° F. for ordinary white metal), long before the main journal bearings, which have behind them the heat capacity of the whole of the crankcase.

With regard to wear, in most lubrication systems the oil is delivered first to the main journal bearings, and passes thence to the crankpin bearings, with the result that the journal bearings receive and retain most of the grit in the lubrication system, and so the more readily lap the shaft.

In general, the rate of wear appears to be almost directly proportional to the load factor on the bearing, and the surface hardness of the shaft, but is almost always most rapid at the bearing which receives its lubrication first and so retains most of the grit.

Connecting-rods.—*The chief considerations in the design of a* connecting-rod are:

(1) That it shall be stiff enough to resist not only bending and crushing, but also vibration.

(2) That it shall be as light as possible.

(3) That the big-end eye of the rod shall be sufficiently rigid to ensure adequate support to the bearing.

Adequately to fulfil conditions Nos. 1 and 3 means in effect that the scantlings of the rod must be such that it has an ample margin of safety so far as ultimate strength is concerned. When I. section rods are used, and this is probably the most desirable form both from the point of view of manufacture and of the disposition of the material, the section must have sufficient width to resist vibration in the plane of the crankshaft, as well as sufficient depth to resist bending—this was at one time too often forgotten and many mysterious troubles and noises were probably due to sideways vibration of the connecting-rod.

It is clearly most important to keep the weight of the connecting-rod as low as possible consistent with fulfilling the other requirements, and in this connection it should be emphasized that it is the weight of the rod as a whole, and not that of the reciprocating end only, which has to be considered. In many cases it is quite as important to keep down the weight of the rotating as it is the reciprocating end, for while the reciprocating mass of the rod affects the balance of the engine, its rotating weight is of more importance in so far as it influences the average pressure on the crankpin and crank journal bearings due to centrifugal loading. In a six-cylinder engine in which the reciprocating parts are balanced inherently, rotating weight plays a more important part than reciprocating; on the other hand, in a four-cylinder engine in which the secondary disturbing forces due to the inertia of the reciprocating mass are cumulative, it is the reciprocating weight of the connecting-rod which must be considered first.

It is in the design of the eye of the big-end that particular care is required.

The first consideration is to obtain a uniform support for the bearing, and, to this end, not only must the eye of the rod be made as rigid as possible in itself, but also the load transmitted down the shank must be distributed over it as uniformly as possible. It is generally useless to provide a wide bearing because of the practical impossibility either of obtaining the necessary rigidity or of distributing the load over it. It is, in the author's opinion, very doubtful whether any useful bearing surface can be obtained when the width of the connecting-rod big-end bearing exceeds three-quarters of the diameter of the crankpin; in any case it is very desirable to splay out the outer webs of the shank at their junction with the big-end eye, in order to distribute the loads transmitted down them.

Another frequent source of weakness in connecting-rod design is the lack of provision of sufficient abutment for the two halves of the connecting-rod bearing. The cap of the rod should be considered as though it were an arch loaded at its centre, but in tension, not in compression. Viewed in this light it is obvious that unless the arch has a wide abutment it will tend to close in and so nip the crankpin at the sides. In high-speed engines it is common practice, in order to save weight, to cut down the width of the abutment to the lowest possible limit, and many failures of big-end bearings are directly attributable to this cause. Fig. 82 shows a design of connecting big-end used in some aero-engines which gave rise to constant failures of crankpin bearings. Fig. 83 shows the same rod after the design had been corrected, when no further trouble arose.

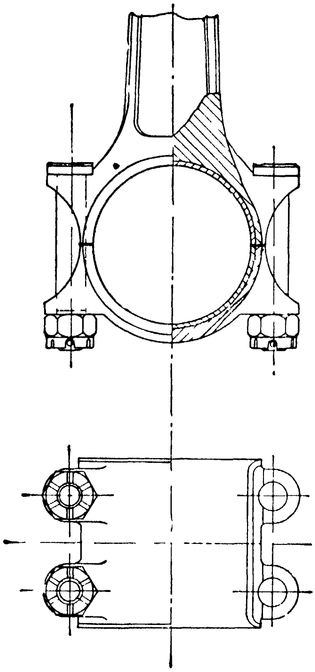


Fig. 82

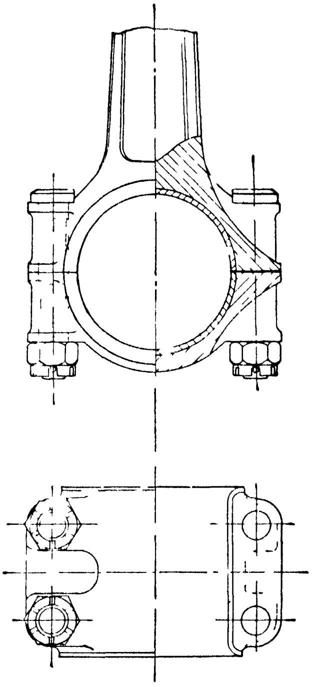


Fig. 83

At one time white-metal was universally used as the bearing lining and gave satisfactory service even in aero-engines. In the modern high-duty internal-combustion engine the severity of the conditions of loading, sliding velocity and heat-flow has, in many cases, gone far beyond the capacity of white-metal. In such cases bearings of other materials are used. For motor-car engines, however, and many other engines of moderate speed and power output, white-metal is still used and is still the most satisfactory material for this purpose. Sometimes an aluminium-tin alloy is used when conditions are rather more severe than the average.

In any event, the white-metal lining should always be kept thin; from 0·010 in. to 0·030 in. appears to be ample. The use of pressed or rolled steel strip backing $\frac{1}{32}$ to $\frac{1}{16}$ in. thick is a way of saving weight without reducing the eye strength of the connecting-rod. In this case a lining of white-metal about 0·010 in. thick is used.

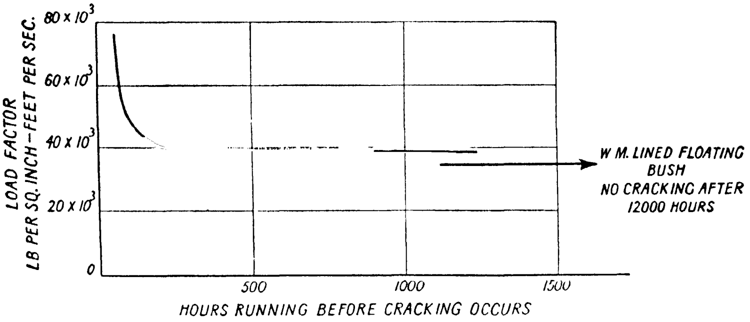


Fig. 84.—Curve relating Load Factor and Hours before Cracking occurs in White-metal Bearings

The limiting peak loading for white-metal is in the region of 2000 lb. per sq. in., operating at a maximum sliding velocity of 18 ft./sec. Beyond the limit of this loading, cracks ultimately appear in the surface, and if running is continued the bearing lining finally breaks up altogether. Fig. 84 gives a curve which relates load factor to life hours to the point of cracking for white-metal split bearings. The curve is derived from a large number of test-bench observations. These values apply to bearing surfaces called upon to withstand the violent rates of pressure change produced by gas loading. For pure inertia loading white-metal will withstand a considerably higher load factor, as is shown by the bearing referred to below.

The cracking process is perhaps best illustrated by giving briefly a summary of the various phases in the life of a bearing which was running near the limit for white-metal. The bearing in mind was of the fixed split type consisting of mild steel shells, white-metal lined to a thickness of approximately 0·030 in., with crankpin diameter 4 in., bearing length 2¾ in., bearing clearance on diameter 0·004 in. The engine, a four-cylinder high-speed 200 H.P. Diesel engine, operated at a governed speed of 900 R.P.M. The bearing load factor for the greater part of its life was 2500 lb. per sq. in. at a sliding velocity of 15·7 ft. per sec.

For the first 800 hours' running the bearing was immune from cracks; at 1200 hours one or two faint cracks discernible with a magnifying-glass had appeared in the centre of the upper shell; at 2650 hours the cracks had become well-defined and a small crazy pattern of triangular and irregularly shaped pieces with sides about ¼ in. had developed; at 3160 hours these cracks had extended over about half the upper shell on either side of the centre and pieces had become loose; at 3700 hours these had become detached, had fallen out and had been wiped round the surface of the bearing. At this stage the bearing was re-whitemetalled.

This short account of the life of a particular bearing shows how, just beyond certain limiting load conditions, alternating flexure of the big-end combined with a high unit pressure on the bearing surface gradually fatigued the layer of white metal, which finally cracked and broke away from its backing.

The behaviour outlined above is typical of a bearing in which the layer of white-metal is run on to the surface of a shell which has been tinned to receive it. The question of adhesion of the white-metal to its supporting material is a controversial subject somewhat perplexing in its uncertainty, but there is no doubt that if good adhesion is obtained the bearing has a much better chance of long life. Many bearings crack after loss of support due to the lining leaving the backing material; other factors, such as deflection under load and distortion due to heat flowing down the shank of the rod, are, however, contributory causes of cracking of the lining, and all must receive careful treatment in a well-designed big-end bearing.

Apart from its inability to withstand high loadings and its low melting-point, white-metal is an excellent bearing material, giving a low rate of wear by reason of its capacity to receive within its surface minute particles of abrasive matter which would otherwise wear the shaft. In practice, a normal carbon steel shaft is sufficiently hard to press home these particles out of harm's way, and for this reason and because of its excellent bearing qualities, manufacturers have been reluctant to use other materials which, although *able to resist high loads and temperatures, bring with them their own special problems and disadvantages. Such materials are inevitably harder and less absorbent of abrasive matter. A shaft of correspondingly harder surface must therefore be used to prevent wear.*

In all high-duty engines and in particular in the high-speed Diesel engine, bearing loads and velocities have gone far beyond the strength of white-metal, and after repeated epidemics of bearing failure, manufacturers have been driven to the use of materials such as lead-bronze, cadmium-tin, aluminium-tin alloys, and the like, to provide a high-duty bearing surface. All these materials must be run with a surface-hardened steel shaft.

In big-ends called upon to endure gas loads which are high in relation to the inertia loading, it is good practice to use a composite bearing consisting of an upper half Y-alloy or lead-bronze shell, and a lower shell white-metal lined. Such a bearing has three main advantages:—

(1) The high loading is sustained by the upper half of the bearing.

(2) Minute particles of abrasive material are absorbed by the lower half, and thus removed from harm's way.

(3) The yielding quality of an all white-metal bearing is retained, thus avoiding a " fire up " which may have serious consequences.

In explanation of (3) it should be said that any irregularity in the bearing surface is much more serious for the shaft if the surfaces are hard than if a soft material such as white-metal is used. Hence the presence of a soft lower half gives some degree of accommodation to the bearing. Moreover, if an actual seizure occurs, the white-metal will run out, thus preventing the danger of seriously damaging the crankpin. There is danger, too, of fracture of the crankpin in an all-lead-bronze or aluminium-alloy bearing if seizure due to lack of oil or other causes occurs. In such cases the heat generation is often so great as to stress the crankpin material highly. Cracks then appear in the hardened surface and fracture may occur.

For these reasons, with materials other than white-metal a larger diametral clearance is given to the bearing:

(*a*) to lessen the risk of local hard bearing which may lead to serious antagonism,

(*b*) to provide a greater flow of oil through the bearing to keep it cool.

In view of the unyielding character of harder materials, which is the bugbear of high-duty big-end bearings, there is a tendency to use very thin bearing shells which, in themselves, give some degree of flexibility in operation and are more amenable to any peculiarities in their environment.

As an example of a bearing operating under conditions of extreme severity may be cited the big-end bearing of a high-speed petrol engine, pin diameter 2 in., bearing length 1·75 in., the upper shell of which is Y-alloy and the lower half white-metal lined mild-steel. The peak loading on the upper half was 4450 lb. per sq. in. at a sliding velocity of 26 ft. per sec.; on the lower half 2280 lb. per sq. in. The normal loading conditions of this bearing were 4830 lb. per sq. in., and 1900 lb. per sq. in. respectively at 22 ft./sec. The bearing showed no signs of cracking or deterioration after some 300 hours' running at a load factor alternating between these two conditions of loading.

As regards the small-end bearing there is little to be said except that to ensure rigidity it should be short and of large diameter. If lined with chilled phosphor-bronze this bearing should never give the slightest trouble, nor show measurable wear, provided that the gudgeon pin is stiff enough to resist deflection (which is seldom the case), and is adequately supported from the piston. Troubles with the small-end bearings of connecting-rods are almost invariably due to deflection of the gudgeon pin, which causes heavy local pressure; so long as this pin is stiff enough to afford uniform bearing, a very small area of surface will suffice. As in the case of the big-end bearing, it is quite useless to provide a width greatly in excess of the gudgeon-pin diameter, both on account of the impossibility of providing adequate support to the bearing and also on account of the deflection of the gudgeon pin itself.

So far as material for connecting-rods is concerned, the problem is somewhat the same as in the case of the crankshaft, but there is one important difference, namely, that the material of the rod is not called upon to function as a bearing surface, so that surface hardness is not required. Since to fulfil the requirements as regards rigidity necessitates automatically the provision of an ample margin of safety, there is seldom any occasion to call for high tensile alloy steels. Plain carbon steels or a mild nickel steel, which can readily be forged or stamped, will be found suitable in all but very extreme cases.

Gudgeon Pin.—Lack of stiffness and inadequate support of the gudgeon pin in the piston are common failings in many high-speed engines. Excessive wear and occasional seizure of the gudgeon-pin bearings are still epidemic in some engines, and although the cause is generally attributed to inadequate bearing surface or scanty lubrication, careful examination will almost invariably reveal that the real trouble is deflection of the pin causing excessive local pressures. Although the pressure on a gudgeon-pin bearing is very high, the rubbing velocity is low, and the average load factor is certainly by no means a high one. Given that the pressure is distributed uniformly over the surface of the bearing, the rate of wear and the risk of seizure in this bearing should be insignificant. In the normal design of trunk piston, the gudgeon pin is carried in bosses which are attached only to the walls of the piston skirt, with the result that the fluid pressure is conveyed to the pin at its two extreme ends, for although the bosses themselves may be rigid enough, their attachment to the piston is by no means so. Calculation will show that, in most instances, the deflection of the gudgeon pin under the maximum fluid pressure is altogether excessive and quite sufficient to concentrate the loading on the two extreme ends of the bearing. It is clearly of the utmost importance either that the gudgeon pin shall receive its load from the piston crown at points as near its centre as the connecting-rod bearing will allow, or its diameter shall be such that when loaded at the two extreme ends, there shall be no appreciable deflection. To conform with the latter condition is nearly impossible, since in many instances it would involve the use of a gudgeon pin of such large diameter and weight as to be prohibitive. When, however, the gudgeon pin is loaded at two points, about half the diameter of the piston apart, there is no difficulty in obtaining the necessary rigidity with a reasonable diameter. As a general rule, for engines of normal compression, the diameter of the gudgeon pin should be one quarter of that of the piston, and its true points of support one half the diameter of the piston apart. With such proportions and with a full floating gudgeon pin the working life of this bearing will be almost indefinite, even with scanty lubrication.

Unless the pin be of abnormally large diameter, it is quite useless to provide a wide bearing at the connecting-rod small end, since its provision necessitates spreading the points of support and so increases the deflection and renders the extra bearing surface valueless. In the author's opinion it is very doubtful whether any use can be made of a bearing width exceeding 0·35 of the piston diameter. Given sufficient rigidity, the wear on a gudgeon-pin bearing is extremely small, but, owing to the small angle through which the connecting-rod oscillates, it is also extremely local and tends to wear the pin oval This tendency can be overcome by the use of a floating gudgeon pin, that is to say, by permitting the pin to turn freely both in the connecting-rod bearing and in the piston bosses, so that it will rotate slowly and wear uniformly all round its circumference. The use of a floating gudgeon pin removes also the difficulty of locating it endwise in the piston, which is a serious trouble, and becomes acute in the case of aluminium pistons, in which, owing to their large coefficient of expansion, the gudgeon pin can be tight only when the piston is cold. Fig. 85 shows the mounting and means of location which the author has found the most satisfactory after much experience. In this design the load is transmitted to the gudgeon pin by two heavy webs extending straight down from the crown of the piston as near the centre as the connecting-rod bearing will permit. The gudgeon pin floats freely in the piston bosses, and is located endwise by means of light washers secured by wire circlips sprung into grooves near the end of the pin.

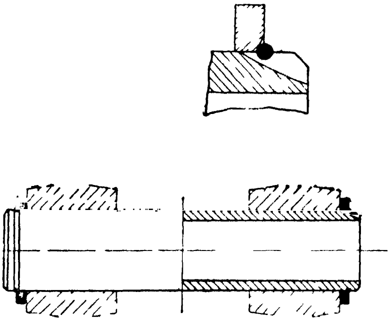


Fig. 85.—Method of locating a full floating Gudgeon Pin

For the material of the gudgeon pin one consideration, namely, surface hardness, should override all others, for if the pin is stiff enough not to deflect appreciably under load, it will be strong enough to resist fracture. From the point of view of surface hardness the best possible material is case-hardened mild steel, and a steel should be chosen which will give a glass-hard surface. In very exceptional cases air-hardened steel may be used, but this is seldom necessary.

Valves.—In the design of the valves, it is necessary always to remember that the objects in view are:

(1) To provide the freest possible entry and exit for the gases.

(2) To keep them as small as possible, consistent with the first condition.

In order to comply with these conditions it is essential to make the orifice coefficient of the valve and its surroundings as high as possible, in order that it shall pass the maximum weight of gas with the minimum of pressure difference. To this end care should be taken:

(1) That the flow of the gases on either side of the valve port is as free as possible—so far as possible there should be no abrupt bends or changes of section on either side of the valve head.

(2) The lift should always be equal to at least one quarter and preferably even to 30 per cent of the port diameter.

For example, the flow of air through an inlet valve may be encouraged to the extent of two or three per cent in volumetric efficiency by giving the whole air passage from the entrance to the casting to the cylinder a slightly waisted venturi form. The general venturi form of the passage immediately under the valve head should be as little interrupted as possible. Also, it is well to provide ample space behind the valve seat for the air stream to straighten up and pass in orderly fashion through the valve. The venturi form should reach its minimum area just behind the valve and should then merge smoothly into the valve seat. The annular passage through the open valve then completes the downstream end of the venturi.

The use of unduly large valves, and more particularly of large valves with a reduced lift, should always be avoided for the following reasons:—

(*a*) Except in very slow-speed engines the provision of relatively large valves must almost invariably be detrimental to the compactness of the combustion chamber.

(*b*) For a given frictional resistance, and therefore for a given volumetric efficiency and fluid pumping loss, a much higher entering gas velocity may be employed when the valve is small, has a high lift, and is well stream-lined, so that the degree of turbulence and therefore the power output and efficiency are greater.

(*c*) Since both the inlet and exhaust valves get rid of the bulk of their heat through their seatings, it follows that the larger the valve the higher will be its temperature. It is very important to keep the temperature both of the exhaust and inlet valves as low as possible. The former because their durability is a function of their working temperature, and the latter because the entering gases are only too ready to take up heat from the inlet valves and so penalize the volumetric efficiency. Heat picked up from the inlet valves may be regarded as purely detrimental, for it is received too late in the induction process to be of any use in assisting uniformity of distribution. Its addition from this source results merely in reducing the density of the charge and raising the whole temperature of the cycle, both of which are highly undesirable from every point of view.

(*d*) Large diameter valves and, in particular, exhaust valves, result in unnecessarily heavy stresses on the valve-operating gear, since on full load the exhaust valves are opened against a pressure ranging from 50 to 80 lb. per square inch. That the valves should always be as light as possible consistent with mechanical strength and heat dissipation is of course obvious, but there is a tendency in many designs of high-speed engine to carry weight cutting in the valves altogether too far, with the result that stretching, distortion, and overheating are liable to occur. When the whole reciprocating mass of the valve and all its auxiliary gear down to the cam are taken into account, it will be found that the weight of the valve head alone forms but a very small proportion of the whole, and it is generally bad policy to stint metal in the valve head and stem for the sake of the relatively small saving in weight effected thereby.

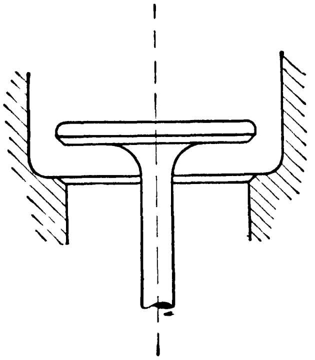


Fig. 86

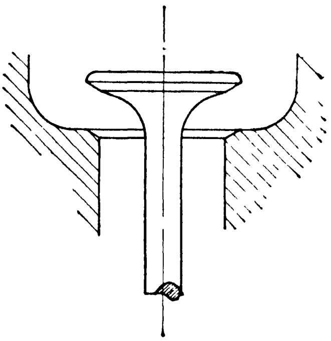


Fig. 87

Since a valve gets rid of its heat largely through the seating, it follows that the seating should be made fairly wide in order to provide a sufficient area of contact when the valve is at rest. Very narrow valve seats save some trouble, both in the first instance and subsequently when grinding in, but there is little doubt that the use of wide seats tends to a higher and better sustained efficiency.

Figs. 86 and 87 show to scale two actual designs of valve and the surrounding walls, both being in the full open position. That shown in fig. 87 is much the more efficient type of the two, and owing to its better stream lining and more efficient orifice coefficient it may be made considerably smaller and, though the head is much more massive, it is for the same weight of gas delivered considerably the lighter valve of the two, and will have much the longer life.

Number of Valves.—In order to keep the temperature of the valve heads as low as possible and so to increase both the durability of the exhaust valves and the volumetric efficiency, it is, in large engines, always preferable to duplicate the inlet and exhaust valves rather than to resort to the use of very large sizes. In the author's opinion, as a rough general rule for more or less normal engines using volatile liquid fuels, no single valve, either inlet or exhaust, should be asked to deal with more than 200 lb. of air per hour, this giving in an engine of average efficiency from 30 to 35 I.H.P. This rule applies nearly irrespective of the speed at which the engine runs. It must of course be considered merely as a first approximation, for there are many other factors which enter into the consideration and are often of such magnitude as to overrule it. First of these is the form of the combustion chamber. When, as is usual in the larger sizes of engine, the valves are placed vertically in the cylinder head, duplication of the inlet and exhaust valves introduces no difficulty and in no way penalizes the efficiency of the combustion chamber. In some cases it is preferable to employ two inlet and three exhaust valves, an arrangement which lends itself very conveniently to an efficient design of combustion head. Such an example is given in fig. 88, which shows the cylinder head and valves of an engine developing 80 B.H.P. per cylinder. In this instance two small exhaust valves and a large one are used, the former being opened some 30° earlier, in order to act as pilot valves to get rid of the bulk of the high-pressure high temperature exhaust products before the large valve opens, and so to relieve this valve and its operating mechanism.

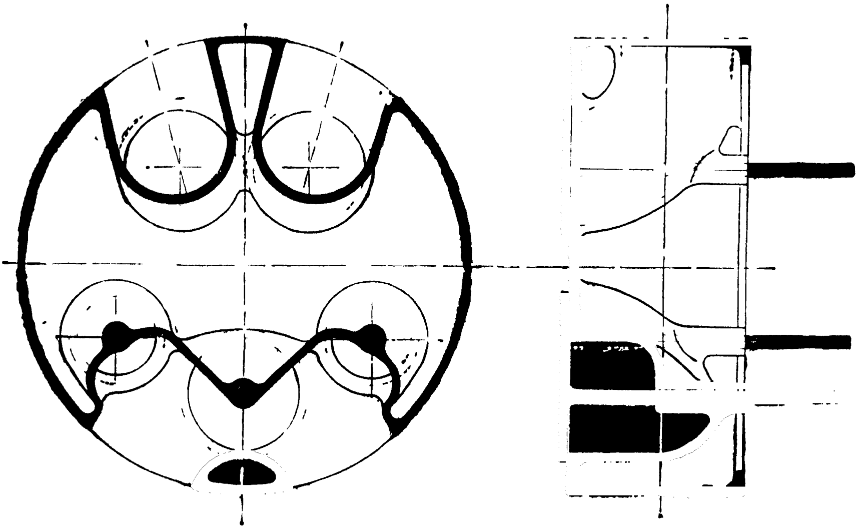


Fig 88

CHAPTER VII

MECHANICAL DETAILS

Ball and Roller Bearings.—The use of ball and roller bearings in internal-combustion engines is becoming more and more extended. The great advantage of such bearings lies in:

(1) Their low coefficient of friction.

(2) Their independence as regards lubrication.

(3) Their freedom, under favourable conditions, from wear.

The disadvantages attaching to them are:—

(*a*) Their high first cost.

(*b*) Their tendency, under certain circumstances, to set up a disagreeable growling noise.

In general, ball or roller bearings should be used:

(1) In all places where adequate lubrication is difficult.

(2) In places where it is difficult or inconvenient to provide a sufficient surface hardness for ordinary plain bearings.

Ball or roller bearings appear to be particularly liable to give rise to noise and lose their efficiency as bearings when applied to any shaft liable to flexure, as, for example, when fitted to the main journal bearings of a very light crankshaft. The efficiency of ball and roller bearings depends on the departure from a purely rolling action, due to deformation of the balls or rollers and their races. The greater the slip, which is responsible for the growling noise of some bearings, the lower the efficiency. Their location on a shaft which is running true, and in as rigid a housing as possible, is of first importance.

Unlike plain bearings, their durability and safety are nearly independent of speed, but are dependent rather upon the *maximum* load to be carried. On the score of reliability, therefore, they show to great advantage in situations where the mean load is heavy and the rubbing velocity very high, i.e. where the " load factor ", as opposed to the maximum load, is very high.

Fig. 89 shows a typical example of a ball journal bearing, and fig. 90 a similar roller bearing.

In the author's opinion a suitable situation for ball, or preferably roller, bearings, is in the connecting-rod and main crankshaft bearings, because, in the first place, these are very heavily loaded, and, at the same time, since the shaft cannot well be case-hardened all over, it is practically impossible to provide the requisite degree of surface hardness to eliminate wear. In the second place, these bearings, and these alone, of all the bearings in the engine, account for a considerable amount of friction, since their load factor is necessarily very high. The use of ball or roller bearings here will serve,

Fig 89.—Hoffman Ball Bearing

Fig 90.—Hoffman Roller Bearing

therefore, not only substantially to reduce the friction and so to improve the mechanical efficiency, but it will, at the same time, by reducing friction, tend to keep the lower part of the engine very much cooler an important feature especially in large engines.

The great objection to their use for crankshaft and connecting-rod bearings is an eminently practical one, namely, the difficulty of getting them into position on a multiple throw crankshaft, since such bearings cannot be split and must be threaded over; this entails the use of a light and lanky crankshaft, the one thing which should most sedulously be avoided. With such a crankshaft, considerable flexure is bound to take place, and this by tilting the inner race and so partially jamming the balls or rollers gives rise both to undue wear and to the characteristic growling noise so often associated with ball bearing crankshafts. It would appear desirable, therefore, if ball or roller bearings are to be used, to build up the crankshaft, using very heavy and massive crankwebs shrunk or pressed on to the journals and so reduce as far as possible any tendency to flexure. In the case of single-cylinder engines with inside fly-wheels and built-up cranks ball or roller bearings may be used with particular advantage, for in this case the main crankshaft bearings are shielded from the maximum shock pressures by the inertia of the fly-wheels, while the use of combined fly-wheels and crankwebs permits of the fitting of a readily detachable crank-pin which can be case-hardened and ground, and being detachable, allows of the connecting-rod eye being unsplit. Fig. 91 shows an actual example of such an arrangement as applied to the "Triumph" motor-cycle engine, which has been found particularly satisfactory from every point of view.

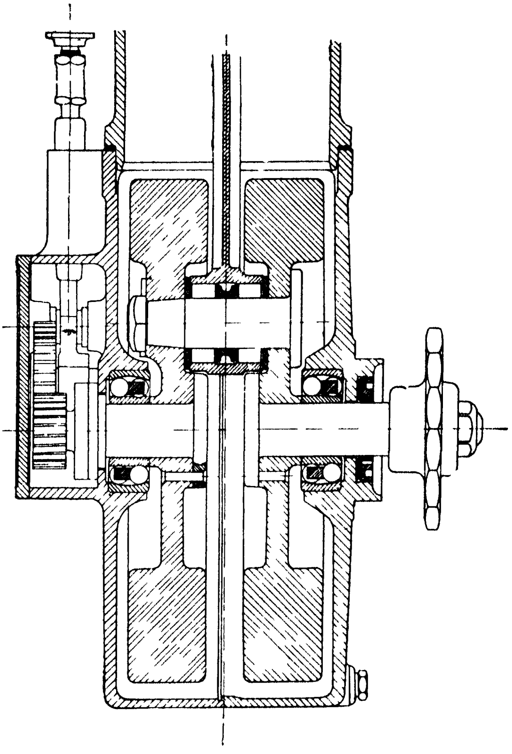


Fig. 91.—Section of "Triumph" Crank chamber, showing Ball or Roller Bearings

Where the loads to be dealt with are heavy, as in the case of crankshaft bearings, it would appear preferable to employ roller bearings despite the objection that such bearings provide no endwise location for the outer races.

This is a serious disadvantage inherent to the roller bearing, but it is probably outweighed by other advantages.

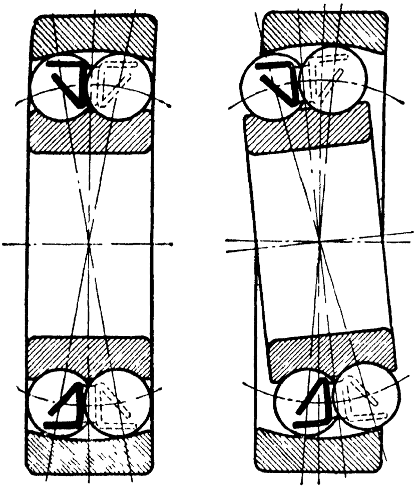


Fig. 92.—Skefko Self-aligning Ball Bearing

Ball or roller bearings should always be so fitted that the race through which the load is first taken shall be a tight fit. In the case of a gear shaft, for example, the inner race should be secured tightly to the shaft either by being pressed on or preferably by being nipped against a shoulder; the outer race should be left a fairly free fit in its housing, and, in the case of ball bearings, should always be permitted a certain amount of sideways float. When a shaft is carried in several ball bearings it is of course essential that a single one only should be used for endwise location by the outer race, all the others being left free to accommodate themselves. Ordinary journal ball bearings are capable of dealing with a considerable amount of side thrust, and, in the author's experience, it is seldom necessary to provide thrust races to deal with casual, as opposed to continuous, end thrust. In the case of roller bearings it is of course essential to provide some independent means of dealing with end-wise location, and it is preferable always to provide ball-thrust bearings for this purpose. In cases where the alignment is uncertain the use of radial ball bearings is to be

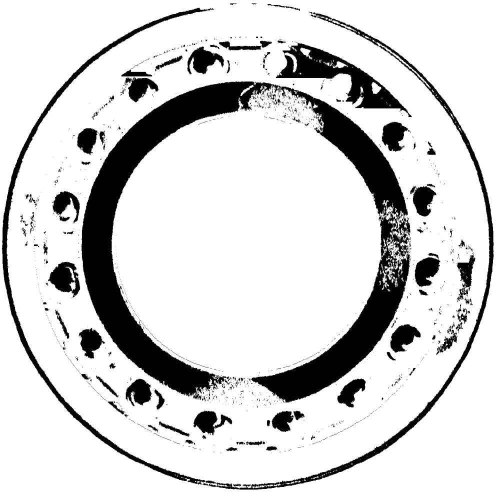


Fig. 93.—Skefko Self-aligning Ball Bearing

recommended, such as the Skefko bearing shown in figs. 92 and 93. Such bearings, however, are liable to give trouble in erection owing to the outer housing slewing round radially when being threaded into position. The author has seen cases where this made it well-nigh impossible to erect certain parts of an engine.

Auxiliary Drives.—In the design of any high-speed engine one of the most difficult problems, if not the most difficult, to deal with is that of driving all the auxiliary gear. This gear consists usually of the camshaft, magneto, oil-pump, water-pump, and in many cases also a fan and a dynamo. It is no easy matter to dispose all these auxiliaries in convenient and accessible positions and to drive them, each at their respective speeds, without noise or undue mechanical complication. From the point of view of silence there can be little doubt that the use of the so-called silent chain is the best means, but it has its limitations. In the first place, it is absolutely essential to provide means for adjusting the tension of the chains if anything approaching a long working life is to be attained. Secondly, it is necessary to provide an arc of contact of the chain with its sprocket wheel of not less than 110° to 120°. This latter consideration precludes the use of a single chain embracing a number of sprocket wheels. At the very outside, a single chain can embrace only three wheels, and this only when the wheels are disposed approximately at the apices of an equilateral triangle. It is therefore generally necessary to provide two chains, an arrangement which is very bulky and cumbersome, and very expensive, for chains and sprocket wheels are at the best of times very costly items. The above remarks refer to the so-called " silent " chain. In more recent years this has been replaced by the multiple roller chain to which the same limitations do not apply, for this type of chain can be wound over and under.

Fig. 94 shows an arrangement wherein a single chain is employed to drive the camshaft and magneto; in this instance the oil pump is driven directly from the camshaft, and there is no positive drive to the water-pump, fan, or dynamo, though the former, if it were required, could be driven from the shaft driving the magneto. In this arrangement provision is made for adjusting the chain by mounting the magneto shaft and wheel in an eccentric housing: the cylindrical portion of this housing is extended through the main wall of the crankcase for a length of about 2 inches, and the magneto bracket and magneto are clipped on to this projection, so that the alignment of the magneto is entirely unaffected by any rotation of the eccentric housing.

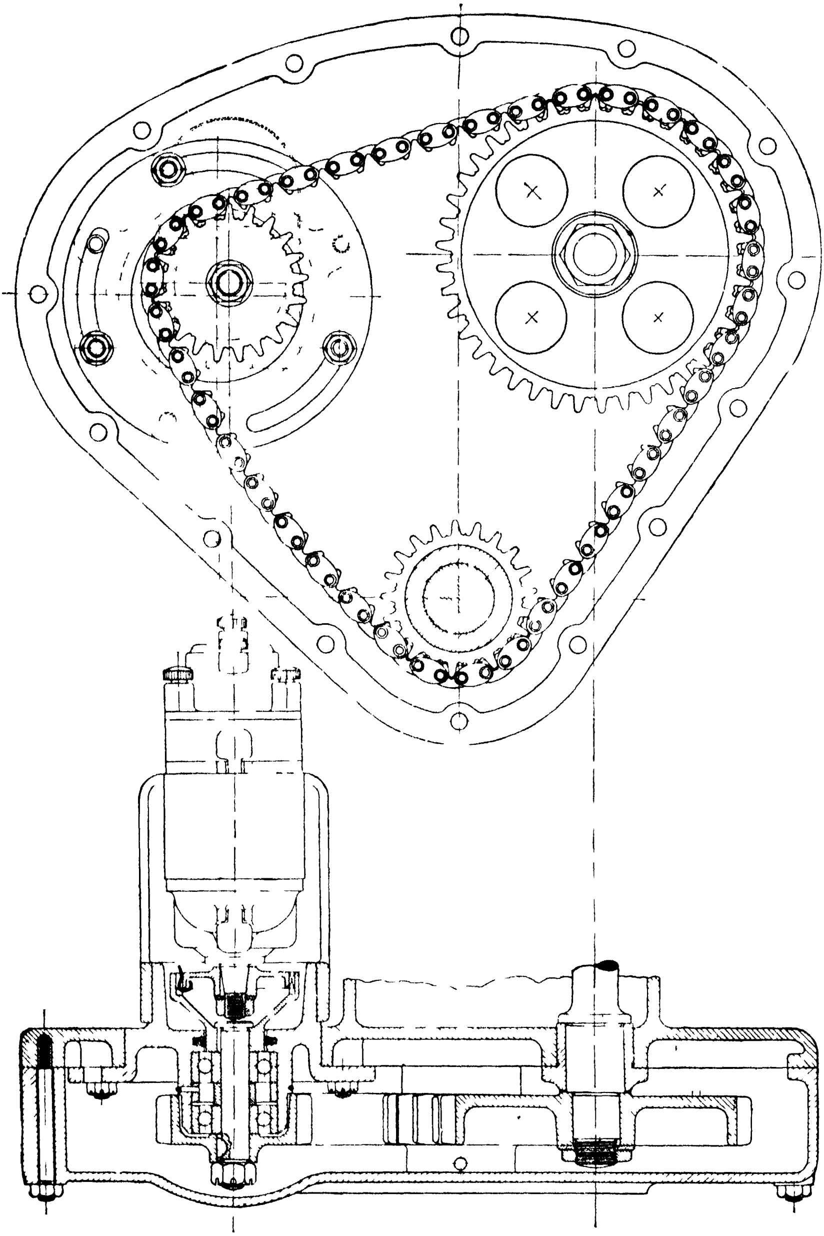


Fig. 94.—Single Chain triangulated Drive with eccentric Adjustment

Fig. 95 shows an arrangement for driving two camshafts, the magneto, water and oil pump. In this arrangement one chain encircles the crankshaft sprocket, a half-speed idle wheel placed immediately above, and a jockey wheel at the side, forming a nearly equilateral triangle. A second chain connects the jockey wheel with a spindle alongside, from which both the water pump and magneto are driven. The oil pump is, in this case, driven direct from the idle wheel and the two camshafts are operated by means of a triangular coupling rod, also from the idle half-speed wheel. This arrangement is very neat and compact, it has the advantage that the chains can be adjusted without disturbing any of the centres of the driven members, and is almost perfectly noiseless, thanks to the short chain centres and to the use of coupling rods for the camshafts.

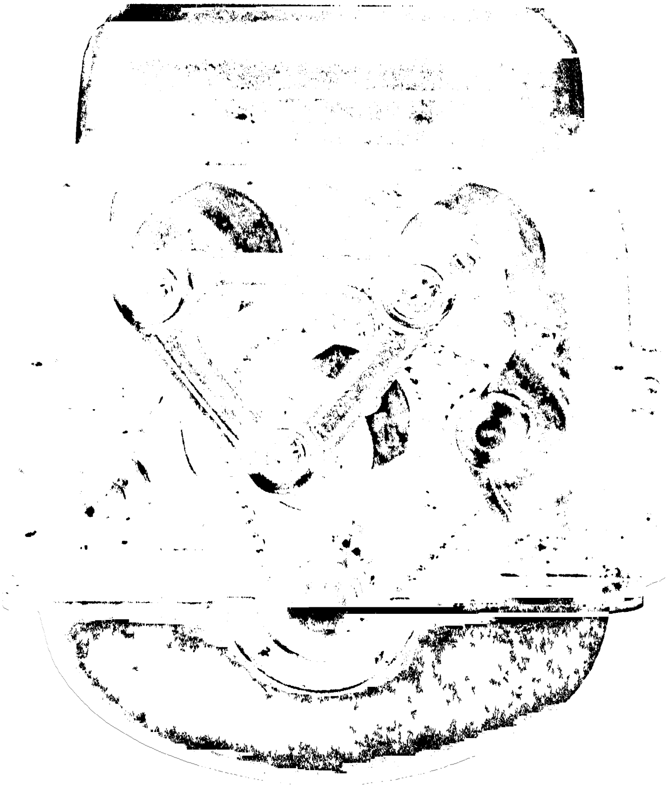


Fig 95. —Method of driving two Camshafts by means of a single triangular Coupling Rod

The great advantage in favour of the use of chains lies in the fact that they act to some extent as dampers for the torsional oscillations of the crankshaft or, at all events, they do not transmit torsional oscillations; whereas, since it requires only a very minute change in angular velocity to reverse the loading on the gear teeth, spur gears have a tendency to chatter and scream when any period of torsional oscillation is passed through.

From the point of view of silence alone, it would be far better, where spur gears are used, to fit these at the fly-wheel end of the crankshaft where the angular velocity is virtually constant. In practice, however, there are generally serious objections to this, both on the score of accessibility and because the shaft is usually provided with a large flange for carrying the fly-wheel, formed integrally with the crankshaft, so that it is not possible to thread the crankshaft wheel into place. Such a position would, however, be vastly preferable on the score of silence, particularly in the case of six-cylinder engines, for the one serious objection to the use of spur gearing lies in its noise, and this is very largely due to variations in the angular velocity of the crankshaft at the end remote from the fly-wheel.

When spur gearing is used, it is of the utmost importance to provide that the wheels shall be correctly meshed. To ensure this it is preferable always to provide some means of initial adjustment, for it is well-nigh impossible to ensure sufficiently correct machining of the centres. When using spur gearing for the auxiliary drives, the author prefers always to mount one or more of the idle wheels in any train in a separate spider carrying the bearings for the wheel and bolted up against the face of the crankcase. This allows of some initial adjustment, for the spider can be attached by bolts or studs having clearance holes, and adjusted until the meshing is correct, when it may be finally and definitely located with a dowel pin.

Fig. 96 shows an arrangement in which two oil pumps and a camshaft are driven from a train of three spur wheels with the intermediate or idle wheel mounted on a spider as explained.

Where the number of auxiliaries is very large, it is often convenient to employ a cross-shaft driven by means of skew or spiral gearing; this arrangement is neat and compact and is much in favour for the drive of the magneto and water pump, both of which can then be disposed in a very accessible and convenient position. Such a drive is satisfactory and silent provided—

(1) That adequate provision is made for dealing with the end thrust involved and for taking up any longitudinal backlash.

(2) That ample lubrication is provided; for such gears, being of the rubbing rather than of the rolling variety, are naturally more dependent upon continuous lubrication.

There are, of course, endless possible combinations and permutations of auxiliary drive, but, generally speaking, the all-spur drive or the combination of spur drive and coupling rod is, in the author's opinion, to be preferred, on the grounds both of reliability and first cost; and provided suitable provision is made for adjusting the pitch correctly, and also providing that the crankshaft of the engine is sufficiently stiff to resist torsional variation, it can be made to run very silently.

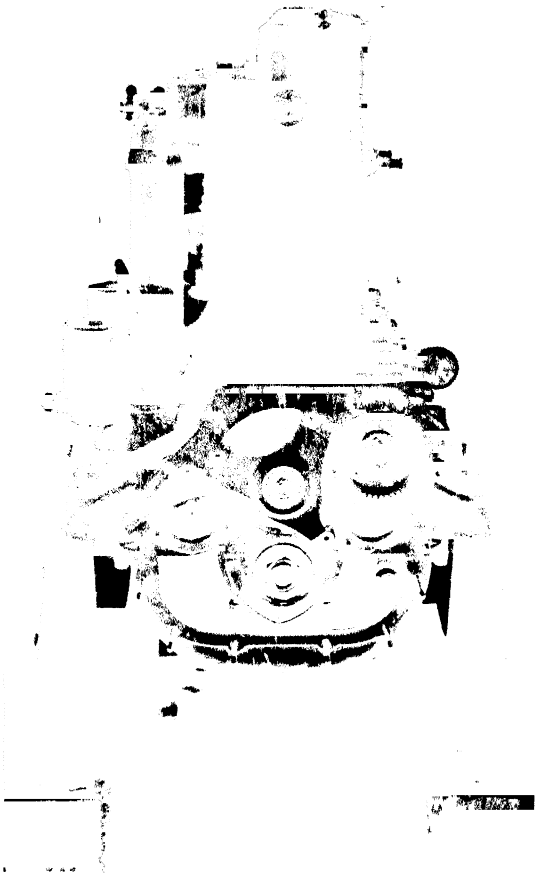


Fig 96.—Photo showing Mounting of idle Spur Wheel and Auxiliary Gear

When applied to motor vehicles a fan and usually a dynamo have also to be driven.

Trouble with fan drives may almost invariably be attributed to attempts to run the fan too fast. It is quite common practice to see the fan belt-driven from the crankshaft of a high-speed engine and geared to run at a speed even in excess of that of the engine. The power absorbed by a fan increases approximately as the cube *of the speed, so that while at 1000 R.P.M. it may absorb only ¼ H.P., at 2000 it will absorb about 2 H.P., and at 3000 nearly 7 H.P.* The belt drive provided is usually nothing like adequate to transmit 7 H.P. at 3000 R.P.M., nor would it be reasonable to expend anything approaching this horse-power in cooling the engine. The practical result is that at any speed in excess of, say, about 1500 R.P.M. the belt slips. This results in rapid wear both of the belt itself and its pulleys. In many cases, if not in most, the normal running speed of the engine is such that the fan belt is constantly slipping, which is highly undesirable, and gives rise to most of the familiar troubles with this part of the mechanism. In general it is far better so to gear the fan that it runs at a considerably lower speed than that of the engine, and so to proportion the belt drive that it will slip only when the engine is running at momentarily excessive speeds.

There are some arguments in favour of driving the dynamo by belt, for the armature of a lighting dynamo has a considerable mass and will therefore tend to run at a constant angular velocity; if it is driven positively from a member which has a varying angular velocity there is bound to be acute disagreement. For this reason the use of any form of toothed gearing for driving the dynamo is undesirable, unless the gearing is situated at the flywheel end of the crankshaft. Chain-driving may be satisfactory, because a chain, thanks to its backlash, its elasticity, and its own weight, is capable of dealing with moderate variations in angular velocity, though a belt is the best of all in this respect.

Lubrication Systems.—Broadly speaking, the various systems of lubrication may be divided into two classes; namely, those which supply the needs of lubrication alone and those which make use of the oil both as a lubricant and as a cooling agent. To the former belong all systems of trough or splash lubrication and those in which a small measured quantity of oil is fed to each bearing, while to the latter belong all those systems in which oil is fed under pressure directly into the bearing itself. Both systems have their advantages and disadvantages, and the choice must depend upon a consideration of all the circumstances. As a broad generalization it may be said that in cases where the load factor on the bearings is high, pressure lubrication is essential for the sake of the cooling effect obtained, while in cases where the load factor is comparatively low, splash lubrication or measured feed may be preferable, on account of the lower rate of wear, since less abrasive grit is imported into the bearing.

In most modern applications, however, the load factor is rarely low enough to permit of the use of the splash or measured feed methods alone. It must be remembered that so long as oil and means of ingress are available, sufficient will always enter the bearing to maintain the oil film, whether it be applied under pressure or not, so that from the point of view of lubrication alone, as apart from cooling, the pressure system scores little or no advantage over the splash.

Pressure Lubrication.—The primary object of using pressure lubrication is to maintain a continuous circulation of a large quantity of cool oil through the bearings, in order to carry away the heat generated by friction. Bearing this in mind, the objects to be aimed at are:—

(1) To circulate as much oil through the bearings as possible.

(2) To keep the oil as cool as possible.

The amount of oil that can be circulated through the bearings of an engine depends upon the pressure at which it is supplied, the clearance in the bearings, and the viscosity of the oil. It should always be remembered that the pressure in itself means nothing; it is only as a measure of the rate at which the oil is being circulated that it has any significance at all. In any pressure system, with a pump of given capacity and normal characteristics as to volume and pressure, the tighter the bearings or the higher the viscosity of the oil the greater pressure will be required to force a given quantity of oil through the bearing in a given time. If now, owing to wear in the bearings or to the use of an oil of lower viscosity, the same quantity can pass through the bearings more freely, the pressure will fall, but this does not mean that either the lubrication or the cooling effect has been reduced.

On starting up, with the oil and the bearings cold, a very high pressure will naturally be required to circulate the oil through the engine, but so soon as the engine warms up the pressure will fall rapidly, owing to the reduced viscosity. Under these circumstances, however, the bearings are being equally well lubricated and in fact better cooled than when the oil pressure is high, since the drop in pressure is due merely to the increased flow through the bearings. This point has been emphasized, because there is a very prevalent belief that, with pressure systems, a low oil pressure indicates defective lubrication, and it is not at all uncommon for operators to use an oil of high viscosity in order to maintain a high pressure in the system, though by doing so they are really both reducing the *flow and increasing the coefficient of friction and therefore the heat generated at the bearings; in other words, defeating their own object.*

In order to ensure a free circulation of oil it is desirable, with pressure feed lubrication, to keep the bearings reasonably slack. From the author's experience it would seem that a minimum clearance of about 0·0015 in. should always be permitted in all crankshaft and connecting-rod bearings when forced lubrication is employed. The practice of putting bearings up tight and allowing them to run in cannot be recommended, since it results merely in both checking the oil circulation and causing undue wear on the crankshaft itself.

Where pressure lubrication is employed and the oil is led into or near the centre of the bearing it is preferable not to use any oil grooves in the bearings, since these merely permit of the escape of oil without compelling it to circulate over the whole face of the bearing and so pick up heat from all parts; it is, however, sometimes desirable to provide flats on the crank to help distribute the flow from the oil hole. Both on the score of reducing friction surface and eliminating the danger of nipping, it is well to relieve away at the sides of the journal bearings, though such relief should not be carried to the extreme ends and so provide a free escape for the oil.

With pressure feed lubrication it is generally desirable to use an oil of low viscosity, since this will permit of a greater quantity being circulated through the bearings in a given time and, by reducing the friction, will reduce the heat generated. So far as the author can discover, the only advantages of an oil of high viscosity are that: (1) the oil film is thicker, the surfaces are therefore kept farther apart, and there is less chance of minute particles of grit bridging the oil film and wearing the metal surfaces; (2) it appears to be less liable to work its way past the piston rings into the combustion chamber, though this latter is very doubtful; and (3) the leakage loss is reduced. From every other point of view all the advantages would appear to lie with the use of a thin oil.

Rate of Circulation.—For engines of high duty the capacity of the pump should be such that, at normal speed, from one to two gallons of oil are delivered per hour for every brake horse-power of the engine. This of course is a purely empirical rate based on practical experience. It does not apply to those cases in which special oil feeds for cooling purposes are provided. In effect it means that even with very slack bearings and with an oil of low viscosity it is about the maximum that can be circulated under a reasonable pressure.

More recent experience with improved oil scraper rings on the pistons and with improved design of oil outlets on the crankpins, has shown that as much as 2½ to 3 gallons per hour per brake horse-power can be circulated through the bearings, and a considerably higher load factor can in consequence be carried. For extreme cases such as racing aero-engines, still further oil-cooling can be obtained by maintaining a brisk circulation of cool oil through the crankshaft itself from end to end in addition to that supplied to the bearings.

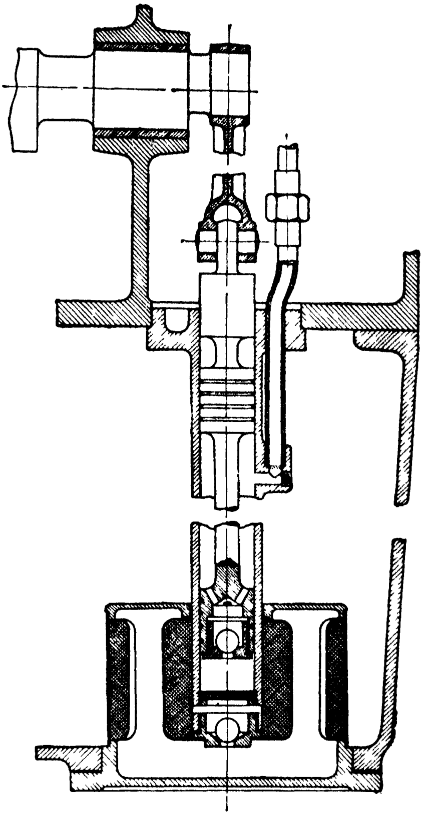


Fig. 97.—Reciprocating Oil Pump

Type of Pump.—There are three types of pump in general use: the ordinary plunger pump, the valveless oscillating plunger pump, and the gear pump. Typical examples of each of these are shown in figs. 97, 98 and 99.

The ordinary plunger pump has the advantage that its volumetric efficiency varies little with wear, and also that it has a high suction lift; but, on the other hand, the suction valve is liable to stick, either open or closed, and so put the pump out of operation; also it limits the speed at which the pump will operate (fig. 97).

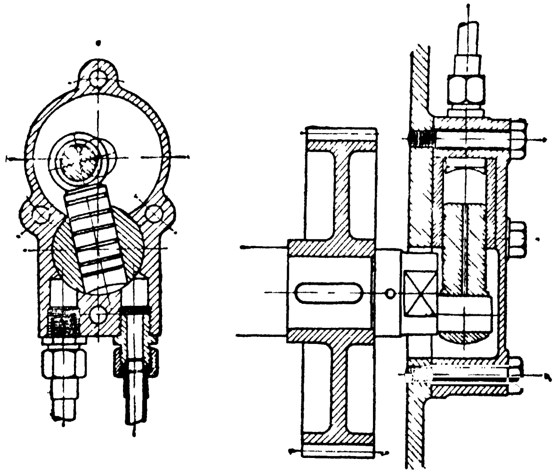


Fig. 98.—Valveless oscillating Plunger Pump

The valveless plunger pump as shown in fig. 98 is, in the author's opinion, more satisfactory than the ordinary plunger pump, for it has no valves to stick or to limit the speed of operation; also it is capable of dealing with dirt, particles of felt, and the other foreign bodies with which lubricating oil is often freely supplied. When run at high speeds it is necessary to provide an air vessel on the suction side in order to maintain a continuous flow in the suction pipe.

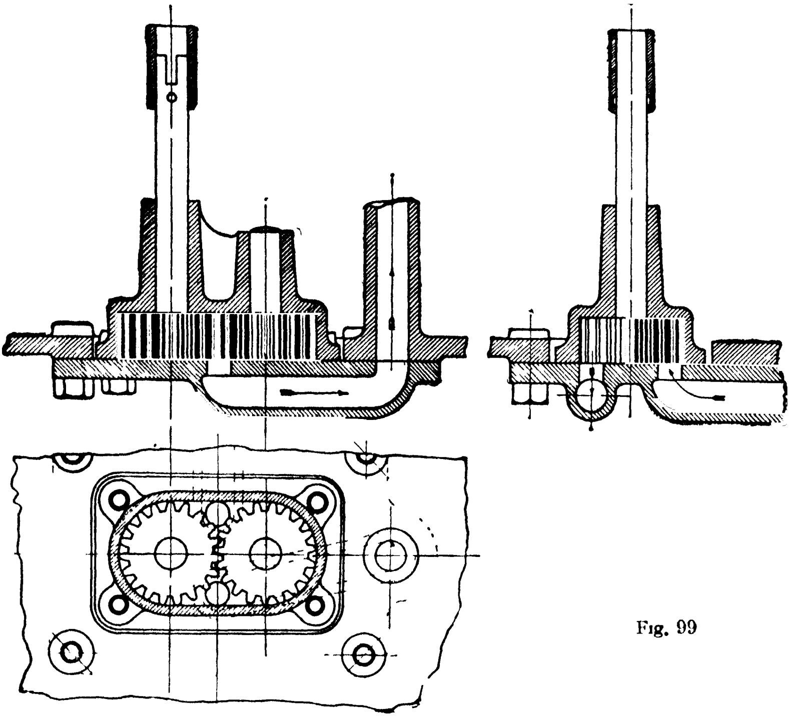


Fig. 99

The gear pump, which is the most commonly used of the three types to-day, has the advantages that its motion is purely rotary, and that it is cheap and easy to accommodate. In earlier days it had four serious disadvantages, which, however, have now been largely overcome:—

(1) Its volumetric efficiency fell very rapidly with wear.

(2) When slightly worn it had little or no suction lift, and unless actually submerged it was liable to let go of the oil.

(3) It was very easily jammed by the entry of particles of foreign matter.

(4) It delivered a very rapidly pulsating flow as the teeth engaged, and this was liable to set up high-periodicity vibrations in the oil piping and so to cause fatigue and fracture of the pipe.

The need in the present-day high-duty engines for a high rate of oil flow, however, gives the gear pump an advantage which overrides these disadvantages. The gear pump can operate at high speed and thus give a high rate of oil delivery from a unit of comparatively small size. Moreover, with the improvements which have taken place in materials and workmanship and in oil cleanliness, the problems of wear, loss of effort, and danger of jamming are no longer so acute, while steel piping may be employed to take care of the stresses set up by vibrations in the oil column. Even so, it is advisable to arrange the pump so that it is entirely submerged, in order to ensure that the oil supply does not fail owing to loss of suction lift.

Oil Relief Valves.—In order to avoid the setting up of dangerous pressures at starting, or when using thick oil or tight bearings, it is necessary in forced lubrication systems to provide a pressure relief valve, and this should be set to blow off at about 25 lb. per square inch. When gear pumps are used and the flow of oil is practically continuous, almost any type of spring-loaded relief valve will serve, but when single-acting plunger pumps are used and the flow is pulsating, the design of the relief valve requires careful consideration. A form of relief valve which the author has found to give very satisfactory results is shown in fig. 100. It consists of a spring-loaded plunger sliding in a cylindrical casing and arranged to uncover relief ports when at the end of its travel. The diameter of the plunger should be from 50 per cent to 100 per cent greater than that of the pump, and its stroke, before uncovering the relief ports, should be nearly equal to that of the pump plunger, so that it has a swept volume equal to two or three times that of the pump. It should be loaded by a long flat-rated spring and will then act as a kind of mechanical air vessel, steadying the pressure at all times, and relieving it when it exceeds any predetermined limit. With such a relief valve the pressure supply from a single-acting plunger pump is kept almost free from fluctuation and will appear practically steady on the pressure gauge.

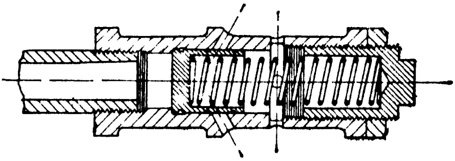


Fig. 100.—Plunger type Oil Relief Valve

Such a relief valve should always be fitted as remote from the pump as possible in order to ensure that the oil has had access to all the bearings before reaching the relief; at the same time it is desirable that the pressure gauge connection be taken from as near the relief as possible, both in order to record a steady pressure and also to ensure that it reads the minimum pressure in the system.

Oil Filters.—The primary objection to pressure feed lubrication is that the circulation of a large quantity of oil through the bearings necessarily involves the circulation also of a large quantity of grit, and therefore tends to more rapid wear. The size of grit which causes ordinary wear, as apart from visible scoring, is such as no ordinary filter can hope to cope with, for it must be remembered that the thickness of the oil film may be of the order of one-tenth of a thousandth of an inch, and that it only requires a particle of abrasive slightly in excess of this dimension to span the oil film and abrade the metal surface. It is obvious that no gauze can possibly restrain the passage of particles of this size, and that the most that can be hoped for is that it will stop the larger particles, mostly of soft material, such as threads of felt from the filtration of the oil, or lumps of carbon from the pistons. Such a filter, therefore, may keep out foreign bodies which are otherwise liable to choke the oil system generally, but it cannot hope to influence the rate of wear.

From the point of view of reducing wear two courses are open to us:—

(1) To provide ample settling space in the oil sump where the fine particles of grit can sink to the bottom undisturbed.

(2) To force the oil through a fine cloth or fabric filter of a fineness capable of dealing with minute particles of abrasive materials.

In addition, it is desirable to provide a coarse-mesh filter, capable of arresting large particles which might choke the system, and one which will not readily become choked itself.

In the first place it is customary to enclose the pump in a coarse filter, usually of gauze, of large area and covering a considerable depth of oil, in order to reduce the risk of choking and so of starving the pump. The filter and pump should be totally submerged and the suction orifice should be arranged either just below the surface of the lubricating oil, so that advantage may be taken of whatever degree of settlement has taken place, or in a small well separated from the main part of the sump by a weir, so that only surface oil is taken by the pump. In the first case the suction pipe may be kept a constant distance below the oil level by attaching it to a ball float and hinging it to the pump intake. When a weir is employed the sump may be made of shallow conical or similar form with a large drain plug at the lowest point of the cone for cleaning out the sediment. The oil pump and suction filter may then form a compact unit in the well. If adequate settling space can be provided, e.g. in industrial engines of such small size as hardly to justify the use of expensive oil-cleaning equipment, a sufficiently high degree of filtration may be obtained by this means.

The same method may be used in road-vehicle engines, but owing to the jolting and changes in inclination suffered by the engine it is almost impossible to obtain a sufficient degree of separation by the settling process, and a fine filter must be used in addition.

For the road-vehicle engine we are thus led to the conclusion that the best compromise is to provide no less than three grades of filtering:

(1) A very coarse mesh filter capable of eliminating nuts, split pins, &c., on the suction side of the pump.

(2) An arrangement such as that already described, by which the pump sucks surface oil, thus eliminating the heavier particles of abrasive material which settle out quickly and are not easily dislodged by vibration or jolting from the bottom of the sump.

(3) A fine filter capable of dealing with minute particles on the pressure side of the pump.

The filter of very fine gauze has the disadvantage that if the mesh is made sufficiently close to deal with minute particles of the order of one or two thousandths of an inch in diameter, the resistance to the flow of oil is increased enormously and an impossibly high pressure is required to force the oil through the gauze. Moreover, it is clear that such a filter would choke quickly and be in constant danger of bursting under pressure.

Other means have therefore to be found to remove the finer particles of grit.

A good deal of experimental work has been carried out on oil separators and filters. One type is the Hele-Shaw "pack" filter in which the liquid to be filtered is passed between the leaves of a series of thin paper or metal discs, threaded on a hollow spindle and packed together in the form of a cylinder. The oil then passes between the leaves and through holes into the hollow spindle whence it is led to the bearings. The advantage of this type is that the most minute particles are detained. It has, however, the disadvantage that for large rates of flow of a liquid of relatively high viscosity, such as lubricating oil, a large "pack" area is required and consequently the whole filter requires a fairly large space for its accommodation, even when used in parallel with the main oil circulation and thus filtering the whole bulk of the oil by degrees.

The filter packs are usually fitted in a cylindrical housing provided with a drain plug for the removal of sediment. The unfiltered residue is left in the annular space between the pack and the sides of the vessel, and the greater part of it settles at the bottom. By arranging that alternate leaves of the pack may occasionally be rotated by hand or, on vehicles, by the operation of the clutch pedal, the particles of grit which adhere to the outer surface may be removed and the efficiency of the filter retained almost indefinitely.

Closely woven cloth materials, such as thick felt made up in cylindrical form, provide excellent filters and remove very fine particles. Unless they are of very large dimensions, however, they are liable to offer too much resistance to be used in series with the main pressure supply and are therefore connected as a " shunt " circuit or " bleed ". In course of time the whole of the oil passes through the filtering process. Such filter elements are discarded and replaced by new ones at intervals. This type of filter has the additional advantages of cheapness, simplicity, and easy removal, and hence is widely used in motor-car engines.

In aircraft engines cloth filters of the type described are often used, but sometimes reliance is placed on a fine mesh gauze filter to remove fine particles from the sump oil. It should be remembered that the air breathed by aero-engines is practically free of the fine dust which pervades the atmosphere at ground level.

The system of lubrication in which a small measured quantity of oil is fed to each bearing has a good deal to recommend it for engines in which the bearings are not heavily loaded and which do not, therefore, require oil cooling.

The quantity of oil so fed should be the minimum required to maintain and replenish the oil film and provide a reasonable margin of safety.

The advantages of this system are:—

(1) Only clean oil enters the bearings, and that only in relatively small quantities, hence the amount of grit or abrasive matter imported into the bearing is reduced to the minimum, with the result that wear also is reduced to the lowest possible limit.

(2) The amount of oil splashing about inside the crankcase is reduced to the minimum, so that the cylinder walls cannot be overlubricated; this is an advantage in one respect, but on the other hand it renders independent lubrication of the cylinders necessary and therefore introduces an extra complication.

The principal objection to this system, and it is a very serious objection, is that the life of each and every bearing so fed is dependent entirely on the continuous operation of the pump or other feeder, and it provides no reserve to tide over a temporary stoppage.

The use of sight-feed drip lubricators and ring oilers may be cited as the simplest expression of this form of lubrication, while, in its more developed form, it is customary to employ either a battery of small slow-moving pumps, each feeding one individual bearing, &c., or a single pump and distributing valve, the principal objection to the latter system being the relatively long interval which must elapse between replenishments.

The system of lubrication shown in fig 101 (p 224), where oil is fed continuously into troughs into which the connecting-rods just dip, is, in its effects, intermediate between the two foregoing. As regards wear, it has the advantage that comparatively little oil is actually forced into the bearings, and therefore the quantity of abrasive introduced is relatively small. It does not, of course, provide for any cooling by oil, since the quantity actually passing through or over the bearings is comparatively very small, but it does provide a good deal of reserve capacity in the event of any accidental stoppage of the pump. One of the difficulties, however, with this system is that it is often troublesome to provide, at all times, sufficient oil for all the crankshaft bearings without over-lubricating the cylinder walls. This difficulty can be met to a large extent by lowering the troughs, and so reducing the depth of immersion, when the throttle is partially closed and the load factor on the crankshaft bearings is reduced. This can be accomplished by hinging one end of the troughs and connecting the other through suitable linkage gear with the throttle lever of the engine, as was done in the Daimler sleeve-valve engines.

When trough lubrication is used, it is sometimes possible to dispense with an oil pump and to use either the flywheel of the engine or a disc mounted on the crankshaft to throw oil up into a gallery from which it may be led down to the troughs by means of suitable ducts. This arrangement has, however, only a limited application, for it is not always possible to provide a disc of diameter sufficient to reach the oil, the highest level of which must be well below that of the troughs. Unless there is room to fit a disc of very large diameter, the permissible range of oil level becomes dangerously narrow.

To sum up: There can be no doubt but that wherever a high duty is required, involving either heavy pressures or high rubbing

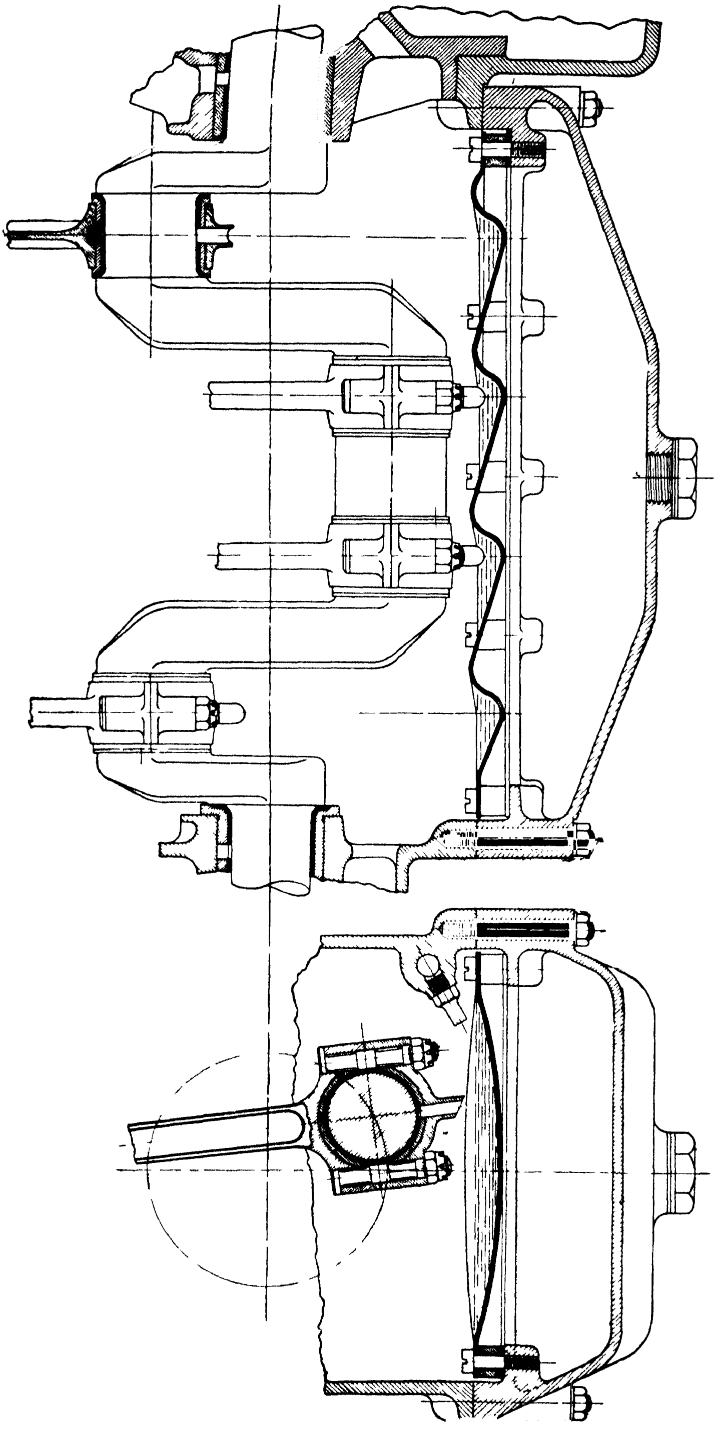


Fig. 101.—Typical Example of Constant Level Splash System

velocities, or both, forced lubrication alone can be relied upon, since this system provides the oil cooling so essential where high load factors have to be dealt with. The one and only objection to forced lubrication is the increased rate of wear due to the large quantity of extremely fine abrasive material introduced into the bearing. This objection can best be met by:—

(1) The provision of ample settling capacity in the base chamber of the engine.

(2) The employment of materials for the bearings giving the maximum possible difference in surface hardness.

(3) The provision of centrifugal filters or separators, though not much is known as yet as to the efficacy of these devices; ordinary filtering, however, is of little use from the point of view of wear.

Measured feed lubrication is no doubt very satisfactory for lightly loaded bearings and gives the minimum of wear, but it involves the use of many oil pumps and provides no reserve capacity; as such it is rather dangerous, since the stoppage of any one pump in the group may prove disastrous. Trough lubrication is, on the whole, a good compromise for engines in which the duty is comparatively light. It tends to show a lower rate of wear than the forced system, though higher than the measured oil system. It has the advantage of providing a very fair reserve capacity, and is on the whole a fairly reliable and satisfactory system once its limitations are realized.

CHAPTER VIII

VALVES AND VALVE GEAR

The timing and operation of the valves are problems which require very careful consideration, for they are factors which have a powerful influence on the performance of the engine, and as such deserve the most careful consideration. Since the timing and opening periods must be decided upon before the cam gear can be designed, it will be well to consider this side of the problem first.

The features to be aimed at as regards the inlet valves are :

(1) To induce the maximum possible weight of charge into the cylinder at full loads.

(2) To expend the least possible energy in the process at all loads.

(3) To produce the maximum of turbulence during the period of entry.

As regards the exhaust valves, the problem is merely that of getting rid of the exhaust with the least possible back pressure and the least distress to the valve gear. So far as the exhaust valve timing is concerned, there is very little to be said except that owing to the high terminal pressure at the time when the exhaust valve is first opened, the velocity past this valve is very high indeed and the heat flow at this period very intense. The high pressure of release, however, usually provides sufficient kinetic energy to counteract the friction and inertia in the exhaust pipe, so that a high mean velocity, both through the valve opening itself and through the ports, &c., is permissible without introducing any appreciable back-pressure, provided, of course, that there is no great resistance imposed on the flow of the gases at the outer end of the exhaust pipe. On the ground of heat dissipation it is desirable, and on the grounds of back-pressure it is permissible, to use small exhaust valves and to work with a high velocity through the valve opening and ports. In practice it is perfectly satisfactory to work with a velocity through the exhaust ports 50 per cent greater than that through the inlet, provided of course that the latter is not already excessive. In a previous chapter it has been shown that, for all-round performance, the best mean velocity through the inlet valves is in the region of 150 ft. per second. The mean velocity through the exhaust valves may, therefore, be in the region of 220 ft. per second.

In the case of the exhaust valves, it is particularly desirable, from every point of view, to use small valves with a high lift, because, in the first place, an exhaust valve can get rid of the bulk of its heat only through the valve seating. The heat carried away along the stem forms only a small proportion of the whole, unless, of course, some expedient such as sodium or copper cooling is employed to convey heat from the head to the stem and thence to the valve guide. It follows therefore that the smaller the diameter of the valve and the greater its lift, the better chance there is of keeping it reasonably cool. In this connection it is necessary to emphasize that, although the maximum area of opening is attained in the case of a flat-seated valve when the lift of the valve is equal to approximately one-quarter of the port diameter, it does not in the least follow that this should represent the maximum lift, because, in the first place, the valve is fully open only for a small proportion of its total opening period, and in the second, the orifice coefficient increases rapidly as the lift is increased; that is, for a given pressure difference a greater weight of gas will pass through a given area of opening when that area is provided by a small valve with a high lift rather than by a large valve with a low lift. Hence, wherever possible, the total lift of an exhaust valve should always be at least equal to, and preferably greater than, one-quarter of the diameter of the port.

Again, the exhaust valve, unlike the inlet valve, has to be lifted from its seat against a pressure which may amount to anything up to 80 lb. per square inch, and for this reason also it is obvious that the diameter should be kept as small as possible in order to reduce both noise and wear and tear.

From every point of view, therefore, it is highly desirable to use the smallest possible diameter of exhaust valve and a high lift.

As regards the timing of the exhaust valve, two factors must be taken into account. It must be opened sufficiently early to permit of the exhaust pressure falling almost to atmospheric before the return stroke of the piston commences, and it must be held open sufficiently late to permit of the residual exhaust gas escaping right up to the very end of the stroke. It is impossible to give hard and fast figures for the most suitable setting for an exhaust valve, because this must necessarily depend upon so many variable factors, such as the mean gas velocity through the valve port and the rate of acceleration of the valve. For a mean gas velocity, however, of about 200 feet per second through the valve port and a normal rate of acceleration, the best setting, in the author's experience, is that the exhaust valve should already have travelled through 50 per cent of its total lift when the piston is at the bottom dead centre and should be still 5 per cent open when the piston is at the top centre. The actual point at which the valve leaves and returns to its seating is practically meaningless as a guide to the valve setting.

With regard to the inlet valve setting, we have to take into account a number of factors which need not concern us in the case of the exhaust valve. Also we have to be much more careful, because not only does the volumetric efficiency, and therefore the power output, of the engine depend very largely upon the inlet valve setting, but also the negative work during the suction stroke may be unnecessarily high. We have to consider how many cylinders are drawing from any one source of supply and also how far our efforts to obtain maximum power output should be subordinated to the attainment of economy on reduced loads. It will be best to consider first the conditions as they apply to a single-cylinder engine for full power, and, later, to note what modifications are necessary to meet other conditions. As in the case of the exhaust valves, we ought to use a relatively small valve with a high lift, though for different reasons. In this case we want to obtain the maximum possible turbulence at the minimum expenditure of energy, therefore we require the highest possible orifice coefficient. For full-load running, in particular, we want to get the highest possible charging efficiency, and to achieve this we want to obtain a high velocity in the valve passage during the earlier part of the suction stroke, and to make use of the kinetic energy we have acquired during this period thoroughly to fill up the cylinder towards the end of the piston's stroke. To this end we require to open the valve rather gradually at first in relation to the piston's movement, to keep it as wide open as possible towards the end of the stroke, and to shut it as quickly as possible after the end. This does not necessitate the use of an unsymmetrical cam, as may at first sight appear, but it depends rather upon the angular setting of the cam in relation to the crankshaft. To illustrate the point:—

Fig. 102 shows an ordinary symmetrical constant acceleration valve lift diagram plotted against crankshaft degrees, where 0° represents the inner centre and 180° the outward centre of the piston.

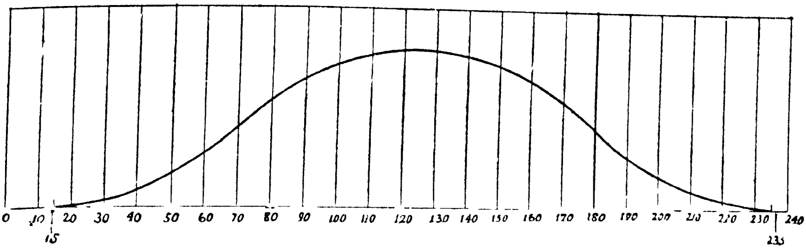


Fig. 102.—Valve Diagram. Constant Acceleration throughout

Fig. 103 shows the same diagram re-plotted in terms of piston displacement, while the dotted line represents the relative velocity

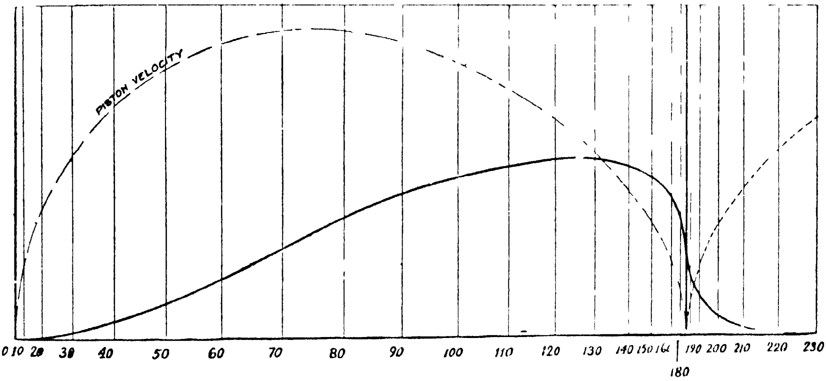


Fig. 103.—Valve Opening Diagram in relation to Piston Displacement

of the piston throughout its stroke (assuming in both cases that the ratio of connecting-rod length to crankthrow is 3·6 : 1).

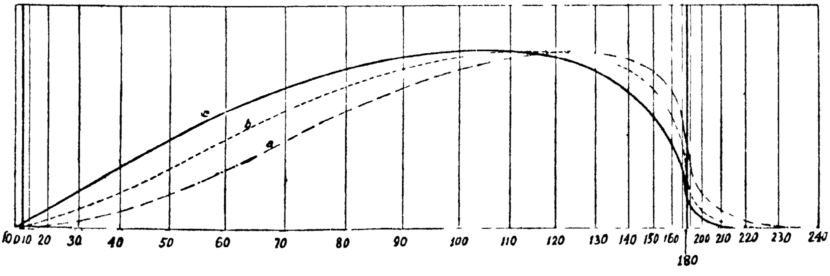


Fig. 104.—Effect of displacing Valve Opening Diagram in relation to Crankshaft

In fig. 104, *a*, *b*, *c* show the effect of displacing the angle of the camshaft by ten crankshaft degrees in each case, and illustrate very clearly how the general form of any given valve opening diagram varies with the angular relation to the crankshaft.

Fig. 105 shows the change of velocity through the valve throughout the stroke in the case of diagram fig. 103, and assuming, for simplicity, that the pressure is constant or that the fluid is a liquid instead of a gas. From this diagram it will be seen that between the middle and the end of the stroke the velocity falls from 300 feet per second to nil, while the kinetic energy, due to this change in velocity, is made use of to pile up a static pressure in the cylinder. With careful design and with a mean gas velocity of 130 feet per second, it should be possible to pile up sufficient kinetic energy, during the earlier portion of the stroke, to overcome the frictional resistance of the valves, &c., and to charge the cylinder up to full atmospheric pressure by the end of the stroke; it then remains to close the valve as rapidly as possible after the bottom dead centre to avoid loss by expulsion during the early part of the compression stroke. This, then, is the setting for maximum power output, but it is not the best for fuel economy, for two reasons:—

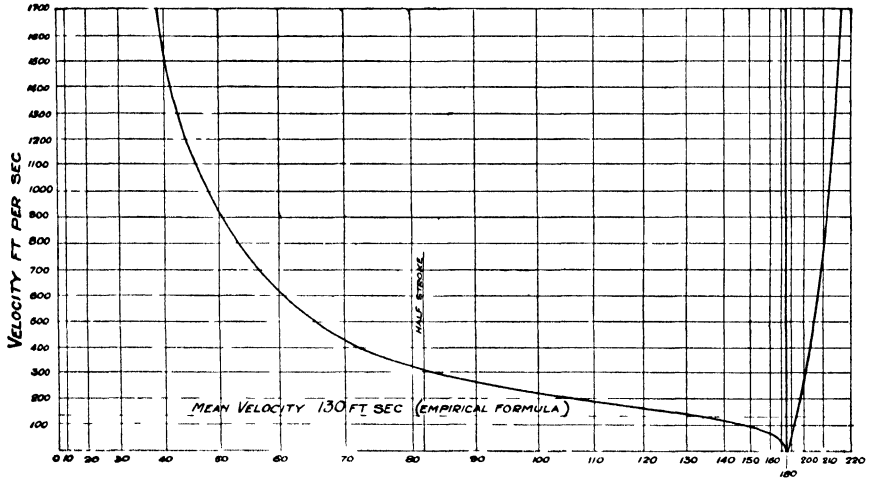


Fig. 105.—Equivalent Gas Velocity through Inlet Valve in relation to Piston Displacement

(1) Work is done by the piston in accelerating the air column during the period when it is travelling at a high velocity, and this is not returned until the piston reaches, or almost reaches, the bottom centre. Consequently the pumping losses are relatively high, though, from the point of view of maximum power output, as apart from fuel efficiency, this is more than compensated for by the increased weight of the charge retained in the cylinder.

(2) When the inlet valve closes and the high velocity flow of gas towards the cylinder is stopped more or less abruptly, a reaction takes place, with the result that the air flows back through the carburettor and some fuel is liable to be blown out and wasted.

If, now, it is desirable to sacrifice maximum power for the sake of better fuel economy, it will be preferable to extend the period of opening of the inlet valve by about 20°, as shown in figs. 106 and 107, to allow it to open considerably earlier and to close a trifle later, the former in order to reduce the pumping losses and the latter to give more time to fill up the cylinder at the lower velocity. In this case, although the opening period of the valve is 20° longer and the actual effective opening area is considerably greater, yet the maximum power output will be slightly less.

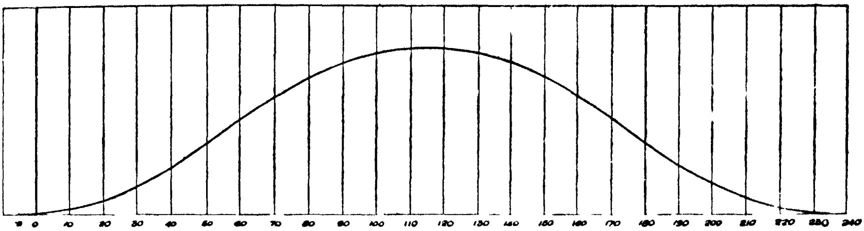


Fig. 106 Valve Opening Diagram with Period extended 20° of Crank Angle

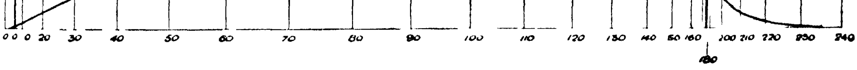


Fig. 107.—Valve Opening Diagram in relation to Piston Displacement

The above considerations hold good only when the carburettor is placed, as it always should be in all single-cylinder engines, reasonably close to the inlet valve port. When any considerable length of induction pipe is interposed between the carburettor and the valve port, pressure oscillations of considerable magnitude will be set up, and these will tend to surcharge the cylinder at certain speeds and to starve it at others, while at all times they will tend to increase the blow-back through the carburettor.

When working on reduced loads, by throttling, we are no longer concerned with trying to fill up the cylinder, and our aim then becomes to maintain turbulence as far as possible and to reduce the fluid pumping losses. So far as the former is concerned, we can only rely on using the smallest possible valves, upon keeping the orifice coefficient as high as we can, and upon ensuring that the gases have as unobstructed an entry to the cylinder as possible after passing the valve. Another point of importance is the position of the throttle. If this is close up to the valve port, so that there is little capacity between the throttle and the valve, then it is clear that during the idle strokes, this capacity will fill up to atmospheric pressure, or very nearly so, in which case the inlet should open early in the stroke and the suction diagram will be as shown in fig. 108; if, on the other hand, there is a considerable capacity between the throttle and the inlet port, then, at the commencement of the outward stroke, the pressure in the cylinder will be approximately atmospheric while that in the port will be considerably below atmosphere, with the result that so soon as the inlet valve opens the pressure in the cylinder will be reduced by gas flowing out through the inlet valve, and there will be some unnecessary negative work on the piston, as shown in the diagram, fig. 109. In such a case it will be preferable not to open the inlet valve until the gases in the cylinder have been expanded down to a pressure corresponding to that in the port; this gives a diagram such as is shown in fig. 110, which should be compared with the previous figure. It is not of course practicable

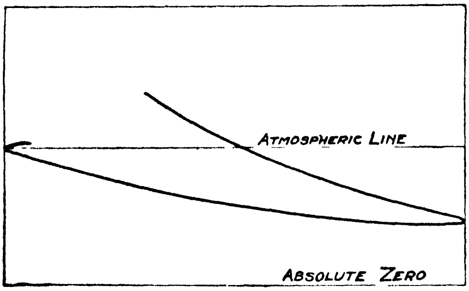


Fig. 108.—Indicator Diagram, Suction Stroke

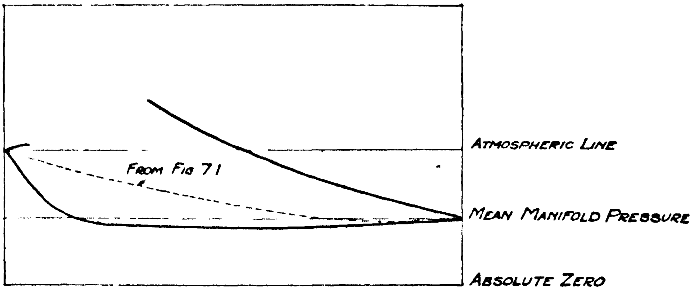


Fig. 109.—Indicator Diagram, Suction Stroke

to obtain a valve timing which will be ideal for all conditions of load, and the best that can be done is to arrange the timing to suit the load at which the engine will be running for the majority of its existence. Generally speaking, for single-cylinder engines it would appear to be best to use a rather late opening inlet valve and to keep the throttle as close as possible to the valve ; this will give the best results in normal working. Apart from the question of negative work, it is always desirable to reduce the capacity between the throttle and the inlet valve as far as possible, in order to subject the carburettor at reduced loads to a pulsating suction, and thus obtain better pulverization of the fuel, rather than to a continuous suck at a low velocity. Consideration will show that if the capacity between the throttle and the inlet valve were infinite the velocity through the carburettor would be continuous throughout the cycle ; while, on the other hand, if there were no capacity, then the mean velocity through the carburettor during the suction stroke would be just four times as great and the pulverization of the fuel correspondingly better.

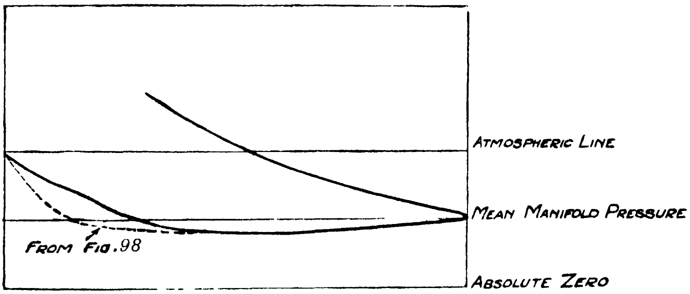


Fig. 110.—Indicator Diagram, Suction Stroke

The case of the single-cylinder engine is relatively simple as compared with that when several cylinders draw from one source of supply. Also it is difficult to treat the problem of valve timing and distribution separately, for they are so closely interdependent; together they form an intensely complex problem, and, probably, the least understood of all problems connected with the modern internal-combustion engine.

We will, however, assume for the time being that we are dealing with a homogeneous mixture of gas and air flowing in the induction pipe, and endeavour to see how best to deal with certain of the more common cases.

Case 1. Two cylinders with cranks at 360° and even firing.

2. *Two cylinders with cranks at 180° and consecutive firing.*
3. *Four cylinders with cranks at 180°.*
4. Six cylinders with cranks at 120° fed by two carburettors, each distributing to a group of three cylinders.
5. Six cylinders with cranks at 120° fed by one carburettor.

So far as the exhaust valve is concerned the conditions are substantially the same in all cases, and it is the inlet valve timing alone which need be considered.

CASE 1. This is simply two single-cylinder engines operating alternately ; there is no overlapping, and the problem is the same as that of the single-cylinder engine.

CASE 2. This is always a very difficult one to deal with. Probably the only satisfactory solution is to provide two separate carburettors and two exhaust pipes and so treat as two separate single-cylinder engines.

When only one carburettor is fitted from which the two cylinders draw consecutively, the best method is probably to employ very late opening inlet valves in order to avoid overlap. It is obvious that if the first piston sets up a high velocity in the induction system and relies upon the kinetic energy so acquired to fill up the cylinder at the end of the stroke, then it is fatal to allow the second inlet valve to open until the first cylinder is completely filled, for the energy acquired will be expended simply in forcing gas into the second cylinder at the commencement of its suction stroke when it is not required, while the first cylinder will be starved. So long as there is any appreciable overlapping in the period of opening of the inlet valves the first cylinder will always be starved while the second will be surcharged. On the other hand, to avoid overlapping and yet keep the inlet valves open long enough to allow the cylinders to fill up is difficult and necessitates a very late opening indeed. Also the short period introduces difficulty in the case of high-speed engines in regard to operation. Again, such very late opening as is necessary to obtain equality between the cylinders will render the suction of the engine very noisy when running on full load, for this is always an objectionable feature of very late opening inlet valves. It would appear that on the whole the best method of dealing with this very unsatisfactory form of engine is, in the case of comparatively slow-speed engines, to open the inlet valves very late ; and in the case of high-speed engines, to employ two altogether separate carburettors and throttles. In either case it is most desirable to fit two separate exhaust pipes, for overlap between the exhaust valves cannot possibly be avoided, and unless two pipes are fitted, cylinder No. 1 will discharge high pressure and highly heated exhaust products into No. 2 just before the completion of the exhaust stroke, thus filling the clearance space of this cylinder with exhaust gas under pressure, at the one point in the cycle at which the presence of highly heated gas is most undesirable. In some instances a fairly uniform power output may be obtained from such an engine because the starving of No. 1 is balanced by the drowning out with exhaust products of No. 2, with the result that on full throttle both cylinders give a much reduced but more or less equal performance, though in such cases the evils no longer balance one another on reduced loads, when the presence of an excess of exhaust products is more than usually objectionable.

CASE 3. Four cylinders drawing from a single carburettor.

Except that the flow in the branch pipe from the carburettor is relatively constant, this case is almost as difficult to deal with as the last. So far as valve timing is concerned, either all overlap of the inlet valves must be avoided, with the resultant difficulties of an unduly short opening period and noise when running on full throttle, or a certain amount of irregularity, coupled with a reduction in the maximum power output due to robbery of one cylinder by another, must be tolerated. In a four-cylinder engine, however, the branch pipes from the throttle to the several cylinders are generally of considerable length, and in some cases use can be made of the kinetic energy in these branches to fill any one cylinder despite attempted robbery by another, more especially so in the case of very high-speed engines. So far as throttled conditions are concerned, this case may be regarded as one in which the capacity between the throttle and the inlet valves is infinite, and, therefore, in which a late opening inlet valve is definitely desirable. On the whole, it would appear that, for this type of engine, it is best to use very late opening and relatively early closing valves in the case of moderate-speed engines and those which operate normally at a comparatively low load factor, because this will tend to give more economical running at reduced loads ; and to use comparatively early opening and late closing valves for engines which are normally run at very high speeds or at a high load factor despite the fact that for maximum power at comparatively low speeds this setting will be inferior to the late opening and earlier closing.

Probably the best results from every point of view can be obtained when two independent induction systems are used, one feeding the inside, and the other the outside pair of cylinders. This arrangement eliminates all question of overlap, and has been applied by the author in the case of several four-cylinder engines designed to give a very high power output and economy.

The indiscriminate use of overlap between the inlet and exhaust valves of the same cylinder is certainly not to be recommended. The arguments for providing overlap in this manner are (1) to make use of the kinetic energy of the gases in the exhaust pipe to scavenge the cylinder and so obtain a greater weight of " live " gas in the cylinder when running at full power; and

(2) To lengthen the period of valve opening, with a view to reducing the stresses on the valve mechanism at very high speeds.

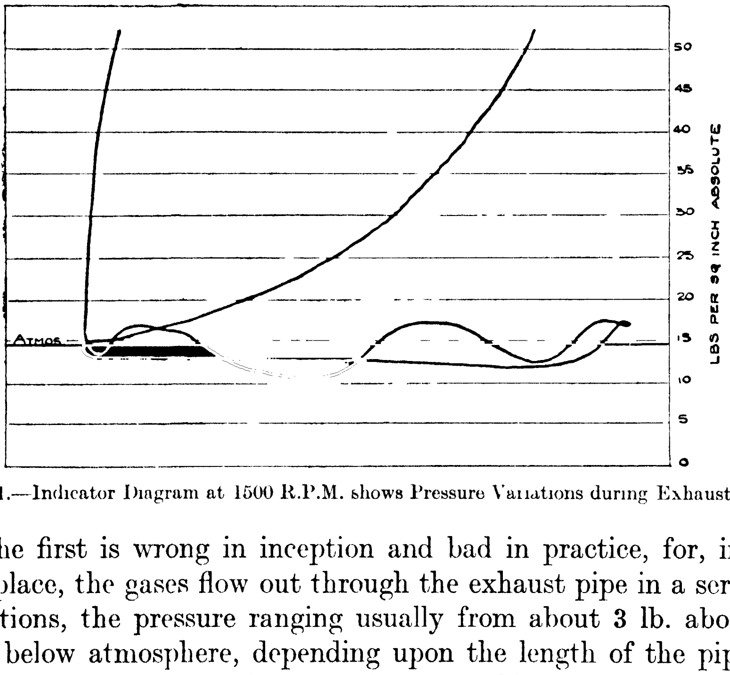


Fig. 111.—Indicator Diagram at 1500 R.P.M. shows Pressure Variations during Exhaust Stroke

The first is wrong in inception and bad in practice, for, in the first place, the gases flow out through the exhaust pipe in a series of pulsations, the pressure ranging usually from about 3 lb. above to 3 lb. below atmosphere, depending upon the length of the pipe, as shown in the indicator diagram, fig. 111, which is taken from one cylinder of an engine running at 1500 R.P.M. It is just as likely that the pressure at the exhaust valve, at the moment when the inlet valve opens, will be above atmospheric as it is that it will be below, in which case a reverse process will occur and exhaust gases will be driven back into the induction pipe. Hence, any advantage which may be gained at one particular speed of rotation will be counteracted by a corresponding but larger loss at other speeds. Again, the valves are seldom far enough apart in the cylinder to permit of any useful scavenging effect being obtained, fresh gas merely being drawn out of the inlet port and into the exhaust, and so lost. On reduced loads the practice of overlapping is particularly bad, for it must be remembered that at this period the pressure of the residual exhaust products in the exhaust system is round about atmospheric, while the pressure in the induction system may be half an atmosphere or less, with the result that exhaust products are simply sucked back from the exhaust system into the induction system, and that under just those conditions when the presence of exhaust diluent is most detrimental to efficiency.

Overlapping is sometimes justified, however, and may be used to give a marked increase in power output:

(*a*) in unsupercharged engines operating over a range of speed sufficiently narrow to permit of the design of an exhaust system suitable to the speed of operation.

(*b*) in compression-ignition engines in which there is no depression in the induction system under ordinary working conditions.

(*c*) in supercharged engines. In this case the introduction of air under pressure overrides any wave effects in the exhaust system. To obtain full response to an overlap timing, however, the exhaust system should not be ignored, and it is well worth while making certain that the exhaust pulsations assist rather than resist the additional scavenging which is the object of the overlap timing.

The second argument need not apply if the valve mechanism is properly designed, as will be shown later. In any event, care should always be taken to prevent adjacent cylinders from exhausting into each other. It is usual nowadays to combine together the exhaust ports of the two central cylinders and to connect the exhaust manifold to the two outside and the central pair; this arrangement is fairly satisfactory, but it is better still to use either three or four quite separate exhaust pipes between the valve ports and the manifold, though in practice this is sometimes inconvenient.

CASE 4. Six cylinders drawing from two carburettors, each feeding one group of three.

This case is simple; there is no overlap of the inlet valve periods, and in so far as valve timing is concerned each group may be treated as three single-cylinder engines.

CASE 5. Six cylinders fed from a single carburettor.

In this case (A) the capacity between the throttle and the inlet valve may be regarded as infinite; (B) overlapping of the opening periods of the several inlet valves cannot, under any circumstances, be avoided; (C) unless long separate branch pipes are provided to *each cylinder—which is almost impracticable on the grounds of distribution, confusion of pipe-work, &c.—little* or no use can be made of the kinetic energy of the gases in the induction pipe owing both to the excessive overlapping and to the constant reversals of direction of flow in the induction manifold. It will therefore be impossible fully to charge any of the cylinders, or, indeed, to attain a final suction pressure in any of them appreciably in excess of the mean pressure in the induction manifold. Since the capacity between the throttle and the inlet valves is so large, it will pay—particularly on light loads—to employ rather late opening inlet valves in order to reduce to the minimum the negative work in the cycle. In such a case the late opening of the valves will cause very little noise at full load because the suction through the carburettor is fairly continuous in any event. Although with six cylinders drawing from one source of supply it is practically impossible to obtain so high a power output as when supplied in groups of three, yet certain advantages may be taken as compensation. In the first place, with such an arrangement, there is much less objection to blow back through the inlet, since any discharge from one cylinder during the early period of the compression stroke will be taken up by another cylinder of the group and will not result in a reversal of flow through the carburettor. It will be worth while, therefore, to allow the inlet valves to close much later than usual, and also to allow them to close much more gradually, so that at low speeds an appreciable proportion of the gas drawn into the cylinders will be discharged back into the induction manifold, while at very high speeds this will not occur, since the suction pressure will be lower and, at the same time, the wire-drawing of the nearly closed valve will be more effective. The practical result of permitting blow back during the early period of the compression stroke will be to reduce the effective compression ratio, leaving the expansion ratio unaffected. The efficiency, therefore, will not be reduced, but the compression pressure at full load will fall automatically as the speed is reduced.

Now, since the tendency to detonate depends in practice very largely upon both the compression pressure and the time element, it thus becomes possible to effect a balance between compression pressure and time. At low speeds, when the time is relatively considerable, the compression pressure will be lower and the tendency to detonate will be equal or less (according to the actual proportions chosen) instead of much greater, as the speed is reduced on full throttle.

It may be argued that, with late closing inlet valves, the useful torque at low speeds will be considerably reduced, but this is true only when comparisons are made with an engine of very low compression and, therefore, low efficiency.

In the case of a high compression engine with early or normal closing inlet valves, full use cannot usually be made of the greater weight of charge retained in the cylinder on account of the excessive detonation set up; it then becomes necessary either to retard the time of ignition, to set the carburettor to give an over-rich mixture at low speeds, or partially to close the throttle. In any case the full torque at low speeds is lost. In effect, the use of late closing inlet valves as applied to such engines is that a considerably higher ratio of compression can be used without trouble from detonation and a substantially higher economy obtained thereby. The actual available torque at low speeds is little, if any, reduced, while the torque at high speeds is increased, owing to the longer period of charging.

The cases considered cover practically all the range; where greater numbers of cylinders than six are employed they are always divided into groups, which come under one or other of the categories we have considered.

To sum up, so far as the exhaust valve is concerned, the problem of its design and operation is the same for all engines irrespective of grouping or numbers of cylinders. It should be as small in diameter as possible; the lift should in no case be less than one quarter of the port diameter, and preferably it should be as much as 30 per cent of the port diameter. In all cases it should be lifted and closed as rapidly as possible, while, as regards timing, it is a good rule that it should be at about half lift at the outward centre of the piston and 5 per cent open at the end of the exhaust stroke.

As to the inlet valve, this, too, should be of small diameter, with a lift approaching one-third of the port diameter, in order to obtain the maximum of turbulence in the cylinder. In the case of a large valve of small lift but of similar area, much energy is expended in eddies at the valve seats, with the result that the final turbulence within the cylinder is reduced. The use of a small valve of high lift improves the orifice coefficient and enables the inlet velocity of the air to be maintained, thus augmenting the turbulence in the cylinder. The opening and closing of the valve must depend, to some extent, upon the number and grouping of the cylinders, but, except in the case of six cylinders drawing from one carburettor, it should always be closed as rapidly as possible, the primary aim being to keep it nearly wide open at the end of the suction stroke and to close it as soon as possible after. With six cylinders drawing from one carburettor, the inlet valve may be opened and closed much more slowly.

Cam Design and Valve Operation.—In designing the cams for operating either the inlet or exhaust valves, the primary considerations are both to open and to close the valves as rapidly as possible with the minimum of stress or noise, and at the same time to arrive at a form of cam which can readily and easily be produced. It is only too often that a cam is designed to give a specific opening diagram, and to provide, say, a constant rate of change of acceleration throughout the whole opening period, which may be ideal on the drawing-board but almost impossible to reproduce with accuracy.

Again, it is always very desirable to avoid any concave surfaces, since these limit the radius of the grinding wheel which can be used to produce them, thus imposing a very tiresome limit on the manufacture. By a suitable combination of cam and follower the necessity for concave surfaces can always be avoided.

In the operation of a spring-controlled valve by a cam, the first movement of the cam imparts a positive acceleration to the valve until nearly half lift, when the acceleration changes and becomes negative—the valve is then under the control of the spring, whose tension must be sufficient to overcome the inertia due to acceleration. From about half lift to full lift, and from full lift to half closed, the valve is entirely under the control of the spring. For the first half of the lift and the latter half of the closing period the spring is inoperative and the movement of the valve is controlled directly by the cam. The spring, therefore, does not come into effective operation until the valve is nearly half open, and ceases to operate when the valve is about half closed. The rate of acceleration permissible while the valve is under spring control is governed by the pressure and rating of the spring, but during the first and last portions of the valve's movement the rate of acceleration is governed solely by the permissible pressure against the flank of the cam. To make the best use of the spring material the rate of acceleration during the spring-controlled period should be such as to correspond as nearly as possible with the rating of the spring—that is to say, the rate of acceleration should increase steadily as the spring is further compressed. During the first and latter portions of the valve's movement the rate of acceleration may generally be much greater, but should be kept more or less constant.

These considerations indicate that constant acceleration throughout the whole period is by no means ideal, or even desirable. The acceleration during the period of cam control may usually be very high and more or less constant, while the acceleration during spring control should be as low as possible in order to use light springs, and should vary uniformly with the rating of the spring. In any case, of course, the acceleration must be limited to that at which the spring pressure will always overcome the inertia of the valve, and that by a margin sufficient to cover any friction of the valve in its guide.

Fig. 112 (p. 242) shows a convenient form of sheet for setting out cam designs. On this sheet are shown:

1. The acceleration, both positive and negative, during the opening period.
2. The corresponding velocity curve.
3. The corresponding valve movement on a time basis.
4. The valve movement in relation to piston displacement, i.e. the valve opening diagram.
5. The evolution of the contour of the cam.

Permissible Acceleration.—This must be considered from two aspects—the highest permissible acceleration while under spring control and the highest permissible acceleration under direct cam control. Both, of course, depend largely upon the total reciprocating weight of the valve and its gear, which must include half the weight of the spring.

With regard to the acceleration under spring control—this is determined by the weight of active spring material and the permissible stress in the material. In the author's experience, so long as the stress in the material is kept down to from 40,000 to 50,000 lb. per square inch, ordinary spring steel coil springs will last almost indefinitely, even in the highest speed engines. When the valve is small, i.e. less than 2·0 inches diameter, and is operated more or less directly from the cam, and when the weight of intervening gear, such as rockers, tappets, push-rods, &c., is comparatively small, a maximum acceleration of about 2500 ft. per second per second, corresponding to a spring pressure of 78 times the reciprocating weight when the spring is fully compressed, is usually permissible, though except in excessively high-speed engines it is seldom necessary to employ so high a rate of acceleration. The acceleration at the point when the spring first takes up the load must, of course, be lower, in proportion to the rating of the spring. In the case of moderate-speed engines there is no need to employ anything like such a high rate of acceleration when under spring control, and for engines of about 15

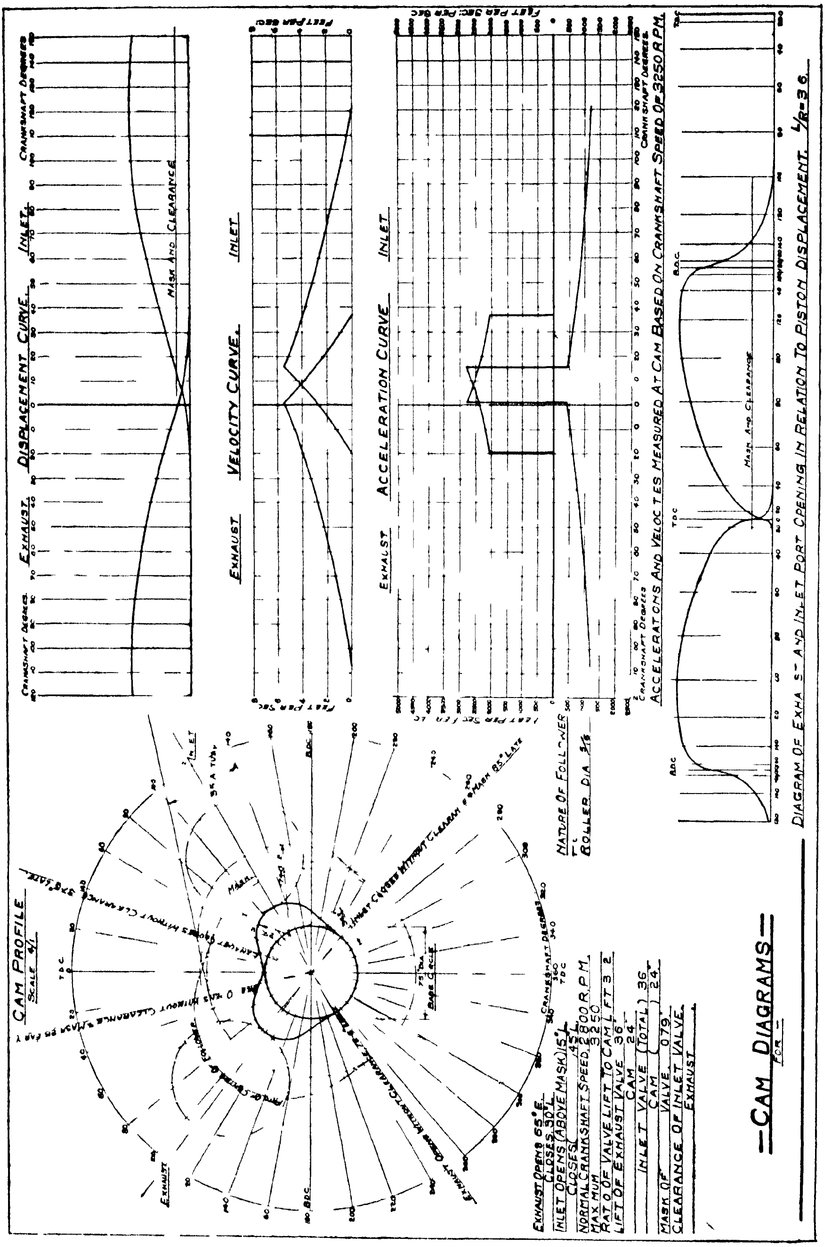


Fig 112 —Cam Diagrams

to 20 B.H.P. per cylinder running at maximum speeds not exceeding 2000 R.P.M. an acceleration under spring control of 800 to 900 ft. per second per second will permit of as favourable a valve opening as can be desired. At the other end of the scale, a limit is fixed for minimum spring tension; this must always be sufficient, in the case of the exhaust valves at all events, to resist the vacuum formed in the cylinder when running throttled. In practice it is found that in order to prevent the exhaust valves from being sucked open, particularly when the engine is in a state of vibration, a spring tension of at least 11 lb. per square inch is required, reckoned on the area of the head of the valve. In the case, therefore, of a valve the area of whose head is, say, 3 square inches, a minimum spring tension of 33 lb. will be required when the valve is on its seating. With a spring of normal rating, the tension, when the valve is fully lifted, will be at least 50 lb. Now the reciprocating weight of such a valve and its attendant gear will probably be in the neighbourhood of 1·5 lb., and the maximum permissible rate of acceleration in such a case will therefore be $\frac{50 \times g}{1\cdot5} = 1080$ ft. per second per second. This, however, makes no allowance for friction in the guide, but even after making a generous allowance for this factor the case for all moderate-speed engines, when the valves are directly operated and the intervening gear is not heavy, is easily met by the provision of a spring of only just sufficient strength to prevent the exhaust valve from being sucked open when running idle.

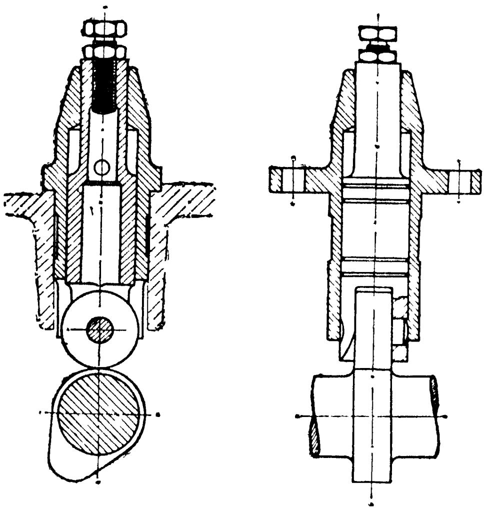


Fig. 113.—Roller Ended Follower

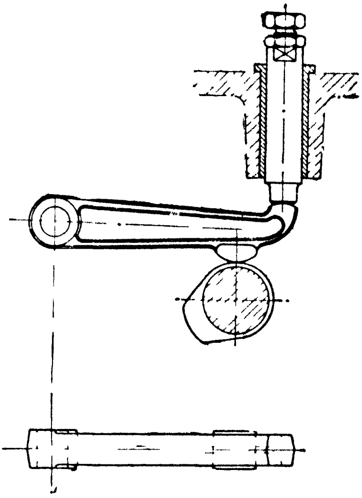


Fig. 114.—Slipper Type Follower

The highest permissible rate of acceleration while the valve is under direct cam control depends upon the type of follower used. This may be either a roller, a curved slipper, or a plain flat-footed or "mushroom"; examples of each of which are shown in *figs. 113, 114, and 115. At first sight, it might appear that the roller is the most satisfactory form, but on investigation it will be* found that this is far from being the case, and for three reasons:

(1) The whole of the load is taken on the roller pin, whose projected area is necessarily very small; this pin cannot conveniently be pressure lubricated, and its facilities for obtaining replenishment of oil are very poor, hence it is easily overloaded.

(2) Owing to the changes in surface velocity as the cam revolves and to the inertia of the roller itself, it follows that the latter cannot truly roll, but must skid, and that just at the period when the pressure on it is at a maximum.

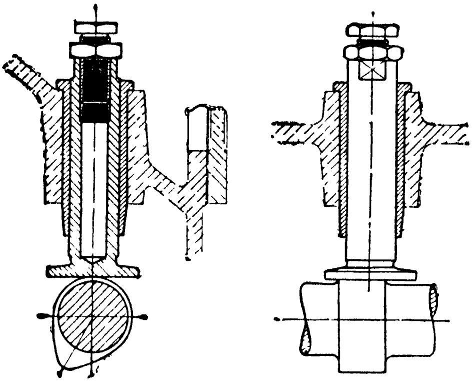


Fig. 115.—Mushroom Type Follower

(3) The use of a roller greatly increases the weight of the tappet gear. For moderate-speed engines, when the loads are comparatively light, a roller is permissible, but not in very high-speed engines.

The second type, namely, the curved slipper, is better than the roller, in so far that it involves no bearing which may become overloaded and break down, but it has the disadvantage that it presents only a very small area of rubbing surface against the cam, and so is liable to wear. Both the roller and the slipper "skid", but the latter skids much more rapidly and presents only one face, while the former skids slowly and presents a continual change of face. Against this, however, must be set the fact that both the radius and the width of the slipper can be much greater than that of any roller. On the whole, the roller has the advantage on the score of wear so long as the pressure is light and the pin bearing is not overloaded, while the slipper scores when the rate of acceleration, and therefore the pressure, is high, for though it may wear considerably and therefore require renewal, it will not break down altogether.

The third type, namely, the flat-footed or "mushroom" tappet, is, in the author's opinion, the most satisfactory of the three, but it, too, has certain limitations, for it necessitates the use of a cam with a larger base circle, which cannot always be provided. Followers of this type should always be offset sideways, so that the sliding of the cam tends to rotate them. Under these conditions practically the whole surface of the flat foot is made use of, and the wear is less than with either of the other two types, while there is no pin bearing to be overloaded. It has all the advantages of the roller type in that it constantly presents a new surface in contact with the cam, and all the advantages of the slipper type, in that it has no bearing to fail, while the conditions as regards lubrication are ideal. A serious disadvantage, however, is the tendency for the foot to bend. In these circumstances high local pressure occurs at the edge of the cam, which digs into and destroys the surface of the follower. This disadvantage can be overcome by making the follower very robust and strong as a cantilever. The other practical objection is that it is usually impossible to employ a low rate of positive acceleration; hence it is difficult to obtain such quiet running as with other types. With flat-footed tappets it is perfectly safe to employ a very high rate of acceleration, for the pressure occurs only when the flank and not the tip of the cam is in contact with the tappet. It is perfectly safe with the latter type, with cams, say, $\frac{5}{8}$ in. wide and a tappet diameter of, say, $1\frac{1}{4}$ in., to apply an average pressure of 250 lb. during the period of cam control. In the case of a valve and its gear weighing 1·5 lb. with a spring tension at rest of, say, 40 lb., this will correspond to a rate of acceleration of about 4500 ft. per second per second. With the roller type it is very doubtful whether it would be safe to exceed an acceleration of about 2000 ft. per second per second. Since, from the point of view of wear, it is only the average pressure which need be taken into account, it follows that there is no particular advantage to be gained by keeping the acceleration constant during the period of cam control, and no objection to following the line of least resistance and making the flank of the cam either a tangent or a simple circular arc.

It is desirable to keep the base circle of any cam as small as practicable in order to reduce the rubbing velocity between the cam and its follower, and this applies whether the follower be a roller or a slipper.

It must be remembered always that the "effective" radius of any cam is the actual radius from the centre of the camshaft to that of the roller or to the centre of radius of the slipper. Hence it matters not, so far as the valve motion is concerned, whether the cam be large and the roller small, or *vice versa*, except when a flat-footed follower is used. In this latter case it is necessary to use a cam of comparatively large diameter, but when flat-footed followers are used the wearing surfaces are so large and the facilities for lubrication so good that a high rubbing velocity is much less objectionable.

The author is greatly indebted to one of his assistants, Mr. R. J. Cousins, for the construction and development of the following method for arriving readily at the most suitable cam contour to comply with any given set of conditions.

When setting out the design of any cam the first question is that of deciding whether the cam profile, as dictated by the valve opening requirements, is permissible mechanically, rather than constructing a cam to conform to some ideal figures for positive and negative acceleration. This being the case, an analysis was made of the general form of cam in which the flanks and nose are composed of circular arcs or straight lines, and a series of graphs prepared, giving practically on sight the actual acceleration at any point for all reasonable proportions of cam.

Fig. 116 shows the acceleration on tangent flanks (dotted curve) and round noses with circular followers, also harmonic cams with flat or mushroom followers.

Fig. 117 gives the figures for hollow flanks with circular followers.

Fig. 118 deals with round (convex) flanks with circular followers.

Fig. 119 shows various forms of cams, internal and external, and indicates the graph to be used in each case.

The formula is the same in all cases, viz.:

$$\text{Acceleration in ft. per sec. per sec.} = \frac{R \times C \times N^2}{100,000},$$

where R = radius in inches (see fig. 119),

C = a coefficient from the corresponding graph, depending upon the form of cam and the angle from the base circle (for flanks) or apex (for noses),

N = revs. per minute or crankshaft (assuming camshaft runs at half-engine speed).

A preliminary lay-out of the cam is first made, taking the known factors (which usually include the approximate base circle diameter, lift, period of opening, and room available for roller or follower).

The positive acceleration at the beginning and end of the flank and the negative acceleration (on the springs) at beginning of nose and apex are then read off, and the form of the curve visualized from the suitable graph, when it will be obvious at once if the cam is out

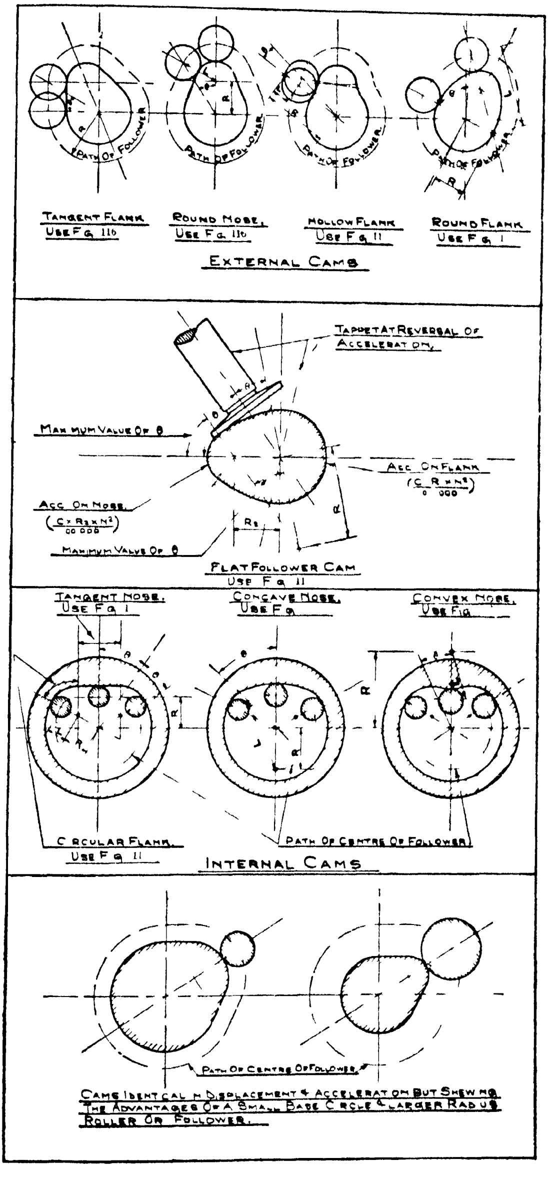

Path Of Follower
Tangent Flank
Use Fig 116
Round Nose.
Use Fig 116
Hollow Flank
Use Fig 11
Round Flank
Use Fig 1
R
External Cams
Tappet At Reversal Of Acceleration.
Maximum Value Of θ
θ
Acc On Flank
$\left(\frac{C\ R \times N^2}{0\ 000}\right)$
Acc On Nose.
$\left(\frac{C \times R_2 \times N^2}{00\ 000}\right)$
R₂
α
Maximum Value Of θ
Flat Follower Cam
Use Fig 11
Tangent Nose.
Use Fig 1
Concave Nose.
Use Fig
Convex Nose.
Use Fig
α
Circular Flank.
Use Fig 11
Path Of Centre Of Follower
Internal Cams
Path Of Centre Of Follower.
Cams Identical In Displacement & Acceleration But Shewing The Advantages Of A Small Base Circle & Larger Radius Roller Or Follower.

Fig 119

of court mechanically, in which case suitable compromises must be made.

Assuming that a tangent cam has been constructed in the first instance, a certain amount of adjustment of the positive acceleration may be made by altering the distance from camshaft centre to the centre of the roller or slipper, the acceleration varying directly in proportion, but if this would make the roller too large on the one hand or make the curvature of the roller or slipper too sharp on the other, the flank may be made curved, concave, or convex as may be necessary, to increase or decrease the positive acceleration.

This will tend to have the opposite effect on the negative acceleration on the nose of the cam, for it will give either more or less time in which to bring the valve parts to rest from maximum velocity.

In cases where the figures are too high all round, it becomes imperative to increase the period to the longest possible and to decrease the lift to the minimum, at the same time providing ample surface on the cam and follower to take the heavy loading.

The negative acceleration on the nose is directly proportional to the distance from camshaft centre to the centre of curvature; it varies also (but not directly) as the proportion is altered between that distance and the radius of curvature measured to the centre of roller or slipper.

These factors provide a ready means of adjusting the negative acceleration, but it is limited inasmuch as a long radius of curvature cannot be used if the cam period is short or the flank convex.

On the other hand, the radius may be shortened down to a figure very little more than the radius of the follower if the engine speeds and lift are low. This may make the nose of the cam concentric with the shaft for a short distance, giving an improved opening diagram.

It should always be remembered that the nose of the cam does not sustain any pressure at maximum speed because the pressure exerted by the springs should then balance the inertia of the valve parts, while, at low speeds, the pressure on this part of the cam must always be somewhat less than the spring pressure. On the other hand, the flank of the cam has to take the pressure of the spring, the gas pressure on the valve (in the case of the exhaust), and also the force necessary to accelerate the valve parts.

It follows that sharp curvature on the nose of the cam will not lead to undue wear or surface cracks, but that the curvature of the cam flank and follower should be as large as possible.

This consideration points to the use of as small a base circle and as large a roller or slipper as possible for any virtual cam, i.e. for any particular line of motion plotted at the centre of the follower.

As the flank will usually be a straight line or a long radius curve, it is affected little if at all by any reduction in the diameter of the base circle, the nose, as explained, is loaded less and less as the speed increases and need not therefore be considered ($\frac{1}{16}$ in. radius will stand quite well), while the roller or slipper is greatly improved by the increase in its radius of curvature, and the rubbing velocity is also reduced.

In all cases of cams and followers formed of circular arcs the velocity and acceleration of the centre of the follower are the same as those of a piston having a crank and connecting-rod of lengths R and L respectively (see fig. 120). Whenever possible the proportions of the nose of the cam should be such that $\frac{R}{L}$ is less than unity.

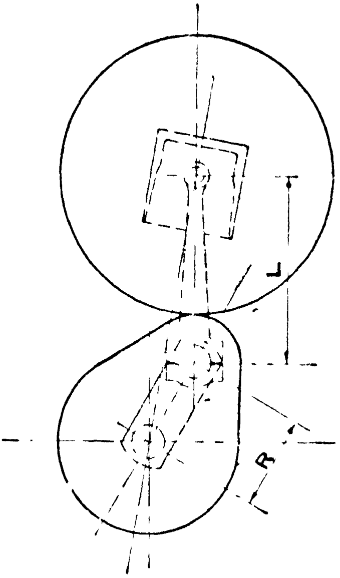


Fig. 120

This ensures that the negative acceleration will be at a maximum at the apex and fall off towards the points of junction with the flanks. Since all ordinary forms of spring give an increasing pressure towards the top of the lift, a cam nose so proportioned permits of the use of a spring which approximately balances the inertia at all points.

If, on the other hand, $\frac{R}{L}$ is greater than unity the acceleration increases towards the flanks, and as the spring must be at least equal to the inertia at any point, it follows that it is too strong at the apex and throws an unnecessary stress on the valve gear.

Flat or "Mushroom" Type Followers.— The cams for use with these are most conveniently constructed of circular arcs, one of small radius for the nose and two others of much larger radius placed symmetrically on either side and tangent to the nose and base circles.

The arcs of the flanks and nose being continued round to form complete circles will be seen to form cranks or eccentrics with which the flat-ended tappet engages in turn. The motion of the tappet is therefore composed of portions of simple harmonic motions of varying amplitude, and the radial velocities and accelerations about the flank and nose are proportional to the distance from the camshaft centre to the centre of curvature in each case.

This is a useful feature of this form of cam, for it enables one to determine a relationship between positive and negative accelerations in the first instance and plot the cam accordingly.

A convenient method of construction is appended (fig. 121).

Construction of Harmonic Cams

(see fig. 121)

Cam profile formed of circular arcs.

Follower made with a flat face to engage with cam and moving in a straight line normal to that face.

Draw a horizontal line AO of unit length, say 1 in.

Draw BO and CO forming angle AOB, AOC, so that each equals half the valve period plus clearance. (Clearance may be assumed to be 12° to 16° total (crankshaft degrees), i.e. 3° to 4° a side actually on the cam.)

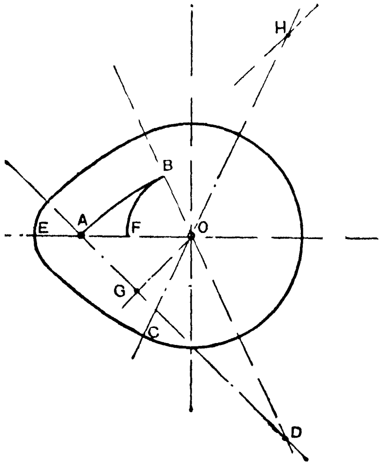


Fig. 121

On BO produced mark off OD, making OD : OA :: acceleration on flank of cam : acceleration on nose (springs). (This proportion may be determined at first or modified after a preliminary lay-out. Average cases range from 2 : 1 to 3 : 1.)

With D as centre, DA as radius, describe arc AB.

With O as centre, OB as radius, describe arc BF.

Then as AF : the required lift :: OB : required true base circle radius. (*Note.*—The true base circle radius is smaller than the cam profile by an amount equal to the radius of the nose.)

Construct a similar figure with true base circle and lift as required. Then take a suitable radius (say $\frac{1}{16}$ in. or more) and describe an arc with centre A forming the nose of the cam, and complete the profile by arcs from centres D, O, and H.

The size of the tappet head is found as follows:—

Join AD and draw OG perpendicular to AD.

Then OG is the maximum eccentricity of the point of contact, and occurs when the tappet is on the junction of the flank and nose curves (i.e. the point of reversal of acceleration from positive to negative).

The radius of the tappet head must be slightly greater than OG, so that the cam may not overrun the edge.

The changes may be rung very easily on the relationship of flank and nose because with this type of cam and follower very heavy positive pressures may safely be used without producing localized wear on the tappet. It will be found that high positive accelerations and long opening periods produce small base circle cams.

Every effort should be made to keep the base circle small, and to this end the radius of the nose may be reduced to $\frac{1}{16}$ in. if necessary with safety. Another convenient property of these simple harmonic cams is that the negative acceleration is greatest at the apex and falls off in a regular manner towards the point of junction with the flank, so that if the spring pressure at the apex be made to balance the inertia at that point and the total deflection from free length to maximum compression be equal to the distance from camshaft centre to centre of nose, the spring pressure will exactly balance the inertia at all other points. This renders the determination of the most suitable spring a very simple matter.

Internal Cams.—These had a considerable vogue at one time on small single-cylinder engines. When composed of circular arcs and straight lines they follow the same laws as external cams of the same $\frac{R}{L}$ proportions, but it should be noted that the actual cam is larger than the virtual cam or line of motion of the roller centre, whereas the ordinary external cam is smaller, so that for any particular case the internal cam has a much higher rubbing velocity.

Moreover, since the cam must embrace the roller or slipper the latter is necessarily very much limited in size and the rubbing speed on the pin considerable when a roller is used.

By the use of very good material and workmanship, and particularly by their success in producing excellent surfaces on the pins and rollers, some manufacturers have succeeded in obtaining very satisfactory results, but the type is certainly not to be recommended from either the theoretical or the manufacturing point of view, the internal grinding alone being sufficient to give the decision in favour of external cams.

The graphs given in figs. 116, 117, and 118, and the key diagrams in fig. 119, are worked out for all reasonable proportions, but the formulæ are given below so that extreme cases may be dealt with. True radial movement of the tappet is assumed in all cases. Where the tappet takes the form of a lever, the fulcrum must be so placed that the arc followed by the roller centre approximates to a radial line, otherwise serious distortion of the valve opening diagram will occur, and stronger springs will be necessary, owing to the fact that the acceleration is increased on one side of the nose and decreased on the other, as compared with the value for a radially moving follower.

The acceleration in all cases corresponds to the second differentiation of the radial displacement in respect to time, and for the regular forms here dealt with is as follows:

Straight line tangent to base circle

$$\text{Acc.} = \text{W}^2\text{R}\frac{(1 + 2\tan^2\theta)}{\cos\theta} \text{ ft. per sec. per sec.} \quad .. \quad (1)$$

Round nose, and round or hollow flank

$$\text{Acc.} = \text{W}^2\text{R}\left(\cos\theta - \frac{n^2\cos 2\theta + \sin^4\theta}{(n^2 - \sin^2\theta)\ 3/2}\right) \text{ ft. per sec. per sec.} \quad (2)$$

Simple harmonic cam with flat follower

$$\text{Acc.} = \text{W}^2\text{R}\cos\theta \text{ ft. per sec. per sec.,} \quad .. \quad .. \quad (3)$$

where R = radius in ft. from shaft centre to roller centre when on base circle in the case of (1) and the radius from shaft-centre to centre of curvature in case of (2) and (3),

W = angular velocity in radians per sec.,

θ = angle moved through from point of contact with base circle for flank (1), (2), and (3) and, in the case of the nose, the angle from the apex = 180° − θ, (2) and (3),

L = radius of curvature in feet,

$n = \frac{\text{L}}{\text{R}}$ (see accompanying figures).

Masked Valves.—It will be evident that as the period required for the inlet valve opening is shorter than that for the exhaust, the accelerations will be greater, increasing inversely as the square of the time. It will also be noted from any ordinary valve diagram that as the velocity of the valve is zero at beginning and end of its period the value of the opening is very small for the first and last 20 or 30 degrees. By recessing the valve seat in such a way that the outer diameter of the valve acts as a piston valve, it is possible to start the motion earlier and finish it later. While keeping the time of opening and closing as before (because the valve head must clear the recess before any appreciable quantity of gas can pass) this greatly reduces the acceleration and usually permits the use of the same cam for inlet and exhaust. It has also a marked effect on the valve opening diagram, since the end of the diagram, instead of being attenuated, retains a considerable value and finishes abruptly.

From the point of view of volumetric efficiency this is the most useful feature of the recessed or "masked" valve, as it is usually called, because it gives a large opening at bottom dead centre and closes the valve before the piston has risen sufficiently to pump the charge back through the valve. In normal cases a mask depth of about $\frac{1}{8}$th to $\frac{1}{10}$th of the lift is suitable, while the total lift should be increased by the mask depth.

Valve Springs.—These must be considered from four points:

(1) The force at various points in the travel.
(2) The maximum stress in the wire.
(3) The stress range from max. to min.
(4) Periodic vibrations in the spring itself ("surging").

(1) From the displacement and acceleration diagrams a force/lift graph is drawn as follows:

A number of points are taken on the displacement diagram from apex to point of reversal (where the nose joins the flank) and are projected down to the acceleration diagram and also horizontally to a vertical line.

On the horizontal lines are marked off the distances A, B, C, D, &c., equal to the corresponding vertical ordinates on the acceleration diagram. These points are joined by a curve which shows the force necessary to balance the negative acceleration in terms of the lift (see fig. 122, p. 254).

Practically all valve springs follow a straight line law, and therefore the straight line which most nearly corresponds to the force/lift graph represents the most suitable spring.

If A is the acceleration at any point in ft. per second per second the force should be $F = \frac{A \times W}{32 \cdot 2}$ plus a small allowance for friction. W is the weight in lb. of the valve, tappet, &c., including half the weight of the spring. (This latter figure may be assumed and a subsequent correction made if necessary.) The minimum force must be sufficient to hold the exhaust valve shut when the throttle is almost closed and to avoid unnecessary loading on the valve gear, the maximum should be just high enough to allow a safe margin beyond the highest engine speed.

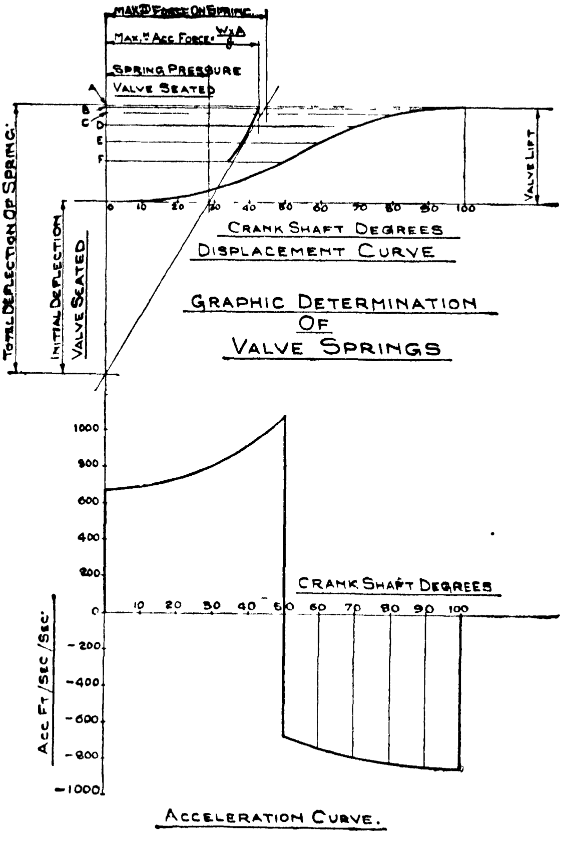


Fig. 122

(2) The maximum stress in the wire must be well below the yield point, otherwise permanent set will take place and the free length will become less, thus reducing the force required to compress the spring to any given point.

It is not advisable to exceed 30 tons per square inch, and it is always preferable to keep within 25 tons per square inch maximum.

(3) The stress range, i.e. the difference between the initial stress (valve closed) and the maximum stress (valve fully open), should be

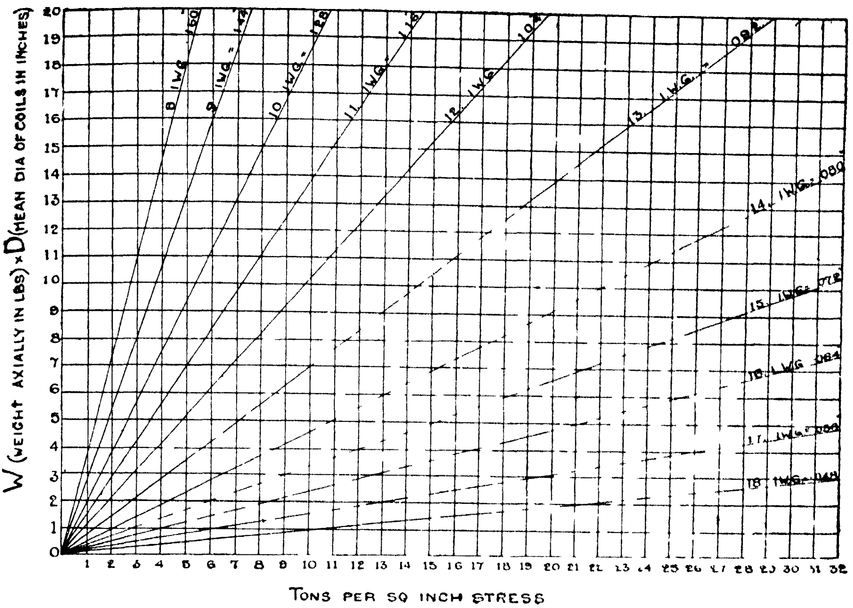


Fig 123 —Spring Graph Stresses

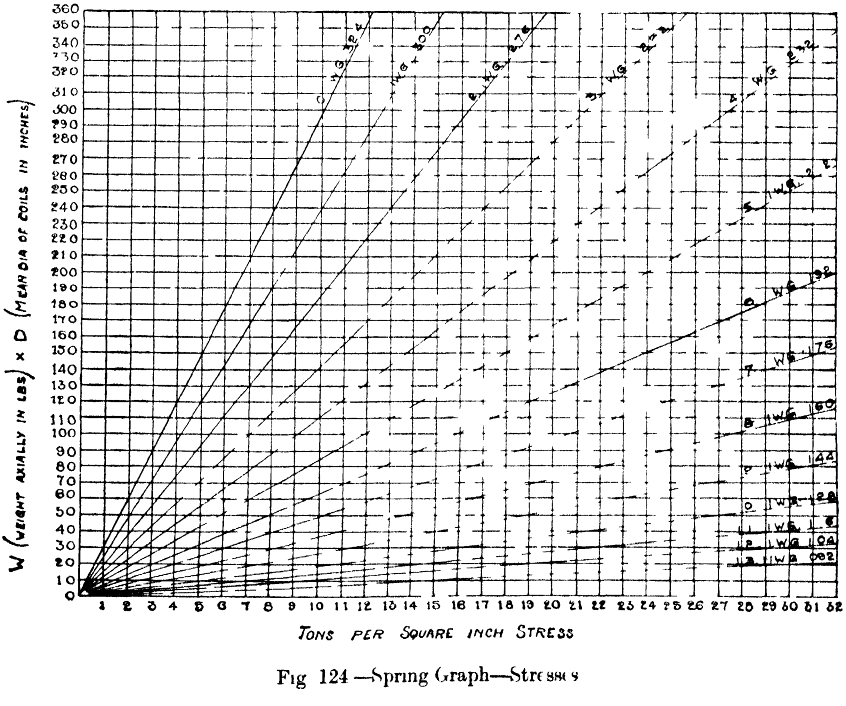


Fig 124 —Spring Graph—Stresses

kept down in order to avoid fatigue and rapid deterioration of the metal of the spring. The range should not exceed 12 tons per square inch according to the quality of the steel and the life expected.

(4) If the mass of the spring itself be too great in relation to its stiffness (which is proportionate to its rate, i.e. the force in lb. required to compress it 1 in. axially) the natural period of oscillation of the spring becomes large and may even approach the period of the valve motion. Serious vibrations may occur, and the spring is liable to fatigue and may allow the valve gear to jump the cam.

The number of free vibrations per minute of the centre of the spring when the ends are held is:

$$N = 590\sqrt{\frac{R}{W}},$$

where N = number of vibrations per minute,
R = rate of spring, i.e. lb., required to compress it 1 in. axially,
and W = the weight of the spring in lb. (active coils only, i.e. weight less dead coil at each end).

If it is found that N is equal to the revolutions per min. of the camshaft or is a simple multiple of same (say 2, 3, or 4 times), it is practically certain that the spring will shudder and cease to function properly.

A vibrating spring appears blurred when the engine is running, whereas in a spring which is functioning correctly, the central coils can be seen clearly owing to the fact that they are stationary while the valve is shut (say two-thirds of the total time) and the eye retains the impression.

The accompanying graphs will assist in the selection of the gauge of wire (see stress graph fig. 123 for light gauges and fig. 124 for heavier gauges) when the force and approximate diameter of spring are known. The deflection per coil (d) may be read off from the deflection graph (fig. 125). The total deflection (D) being already fixed (see figure for force/lift graph) the number of *effective* coils is $\frac{D}{d}$ and the total number $\frac{D}{d} + 2$.

$$\text{The maximum stress} = \frac{8FD}{\pi d^3} \text{ lb. sq. inch,}$$

where F = maximum force in lb.,
D = mean dia. of coils in inches,
d = dia. of wire in inches;

$$\text{also the maximum deflection} = \frac{8FND^3}{Cd^4},$$

where N = effective number of coils (total minus 2),
and C = the transverse modulus of elasticity = 11,600,000.

The spring thus arrived at should then be considered under heading (4) before being passed as suitable.

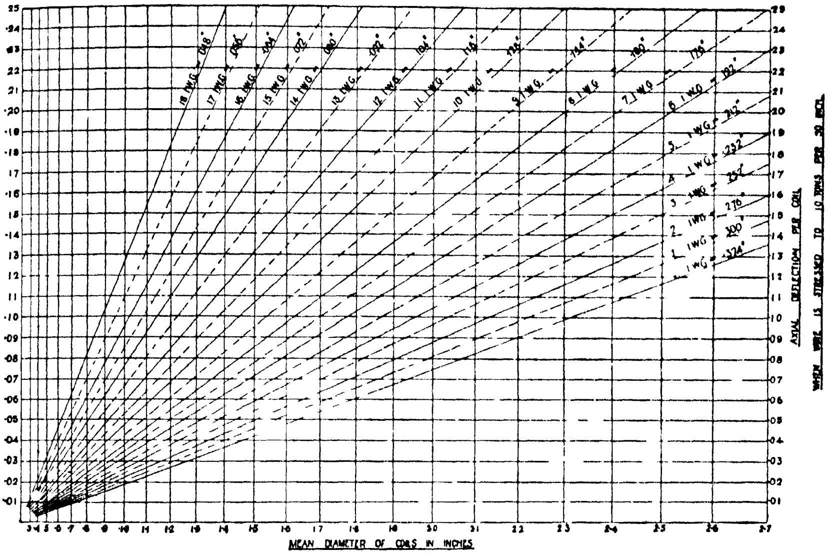


Fig. 125.—Spring Graph-Deflection

It will be apparent that (3) and (4) are in opposition, for if the stress range be kept *too* low, the spring loses stiffness and may develop slow vibrations in time with the valve gear. There is no difficulty in practice in satisfying both points even in very high-speed engines.

If the length of the spring be made such that the coils are almost crowded when the valve is open the stress due to "surging" may be limited.

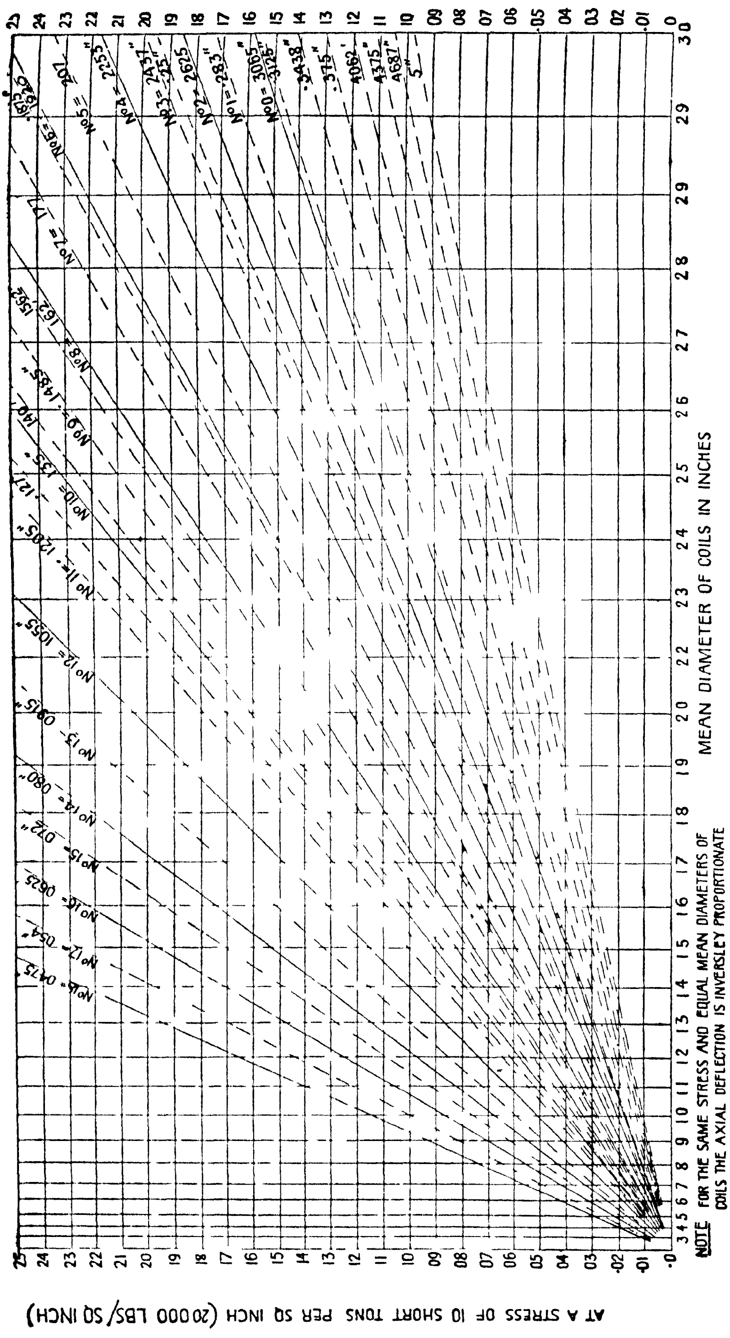


Fig 126 —Spring Graph—Deflection

For American (Washburn & Moen) Wire Gauge

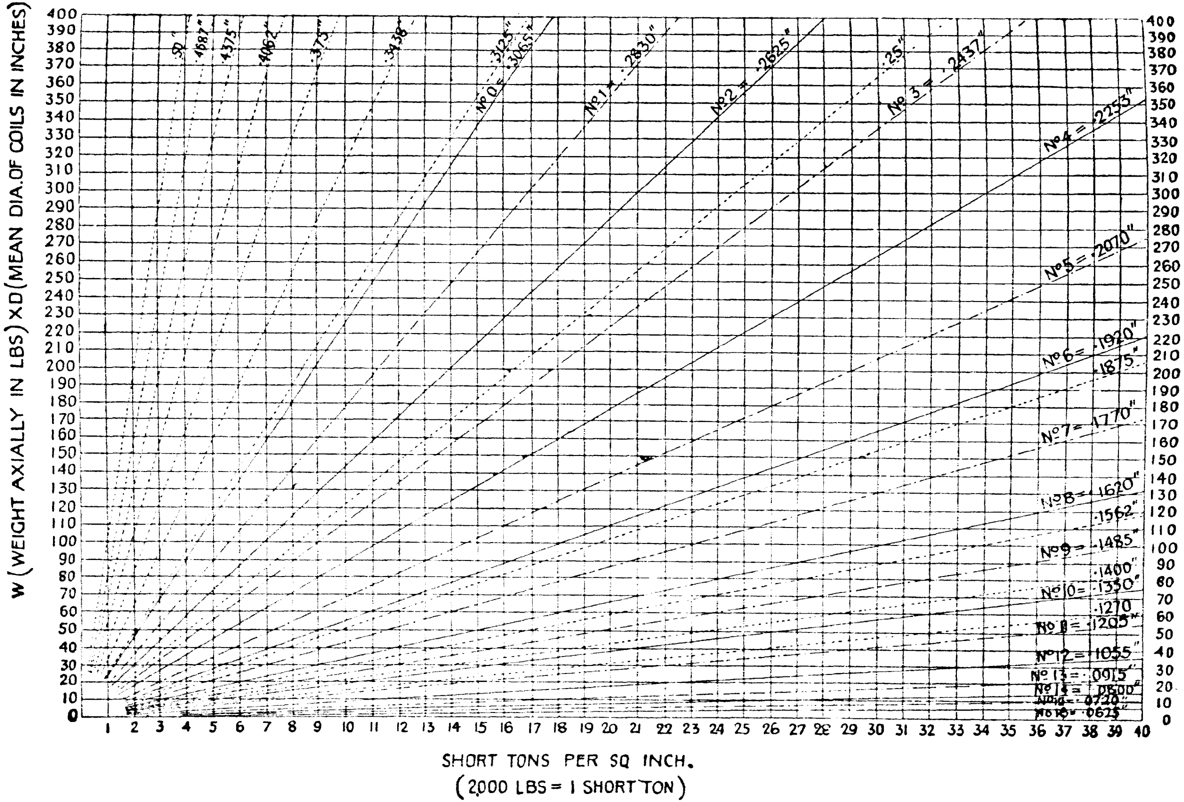


Fig. 127.—Spring Graph—Stresses
For American (Washburn & Moen) Wire Gauge

CHAPTER IX

PISTON DESIGN

Of the total mechanical losses in any internal-combustion engine of normal design, piston friction forms by far the largest individual item; and, of all the independent component parts the piston is probably the most difficult of design in that it has to fulfil so many and frequently such contradictory conditions. Broadly speaking, the main objects to aim at in the design of a piston for the lighter high-speed types of internal-combustion engine are:—

(1) To reduce friction to the lowest possible limit.
(2) To reduce the weight.
(3) To dissipate heat to the walls of the cylinder.
(4) To prevent the passage of oil into the combustion chamber.
(5) To provide adequate support for the gudgeon pin.

Conditions 1 and 2 are largely interdependent, for the bulk of the average pressure exerted by the piston against the cylinder walls is, in any high-speed engine, due to the resolved component of the inertia forces which, when averaged over the whole cycle, exceeds the fluid pressure; hence if the weight is reduced the average bearing pressure is reduced also, and for the same bearing pressure per square inch, and therefore for the same durability, the area of bearing surface may be reduced nearly in proportion to the weight.

Piston friction is of course dependent also upon the nature and condition of the lubricating oil adhering to the cylinder walls.

The general question of lubrication and friction has been considered in relation to bearings, etc., in Chapter V, and it has been shown that friction is to a large extent proportional to the area of surface, the viscosity of the lubricant, and, to a much less extent, to the load. In the case of the piston, however, the conditions are somewhat different; in the first place, although the rubbing velocity is higher, the average load is low and is, compared with any of the bearings, small. Under such conditions the area of surface and the viscosity of the lubricant play a very important part.

With regard to the area of surface, it is clear that only the surface at right angles to the line of the crankshaft is operative, the surface at the sides of the piston receiving no thrust at all. It is therefore clearly desirable to remove all inoperative surface in order to reduce, as far as possible, the area of the oil film in shear. In view of the very light loading to which a piston is subjected, a relatively small area of bearing surface suffices, and for a reasonably light piston an actual bearing surface on either side equal to 50 per cent of the area of the crown should be ample, provided it is properly disposed; that is to say, provided it is disposed equally above and below the gudgeon pin and over a subtended angle of from 90° to 110°. The author has never heard of any case of a piston seizing from overloading of the bearing surfaces. Seventy per cent of piston seizures are due to insufficient allowance for expansion or to distortion, and the remaining 30 per cent to complete failures of the oil supply. As to wear, this is mainly due to the piston rings and apparently does not depend on the piston material; it is very unusual to find any serious wear on the bearing surfaces of a piston or in the cylinder bore below the travel of the piston rings.

When compared with any other bearing surface in the engine, the friction of the piston is abnormally high; this is undoubtedly due primarily to the fact that the oil is partially carbonized and its viscosity, and therefore its resistance to shear, is greatly increased. It must be remembered that, at every cycle, most of the oil clinging to the walls of the cylinder barrel is exposed to the full flame temperature of the burning gases. It is probable also that the fluid resistance is greater when the direction of motion is constantly reversed than when it is continuous, as in the case of a shaft running in bearings. As an illustration of the effect of the carbonizing of the oil on piston friction, the author has always observed, when testing engines on electric dynamometers, that if the supply of fuel be suddenly cut off after running under load and the engine motored round by means of the dynamometer the friction torque is at first high, but falls rapidly as the carbonized oil on the cylinder walls is replaced with fresh clean oil from the lubricating system.

Fig. 128 shows a typical curve of total friction torque on a time basis carried out under these conditions. The engine in this instance was run under full load at 1200 R.P.M. for a considerable period, until all temperature conditions had become normal; the supplies of circulating water and fuel were then simultaneously cut off and the engine motored round at precisely the same speed—the change over from full load running to motoring being effected without any measurable interval of time and without any appreciable change in speed. In this particular case the friction

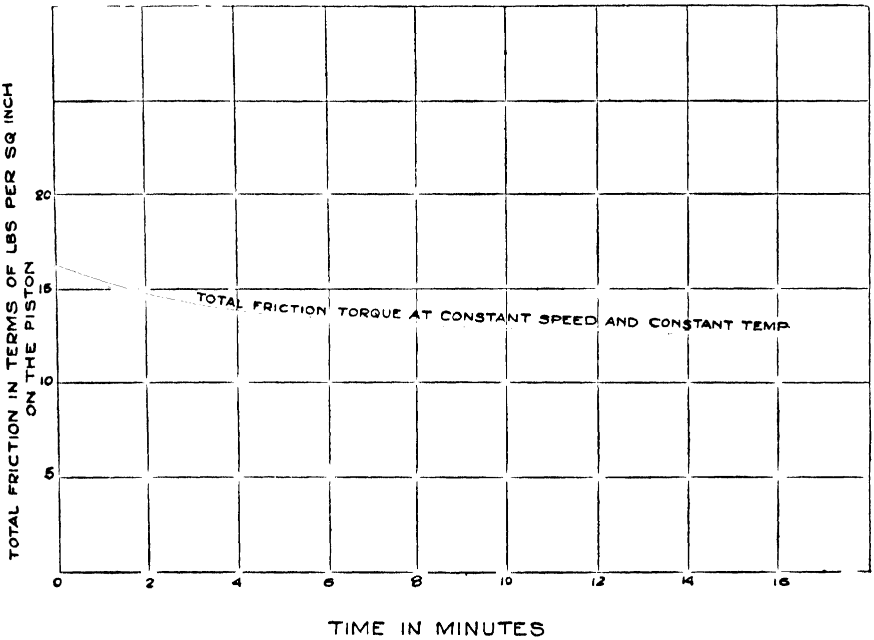


Fig. 128 —Curve showing Drop in Friction Torque as contaminated Oil is replaced by fresh Oil

losses of the bearings and auxiliaries and the fluid pumping losses had all been ascertained separately, and were found to be equivalent to a mean pressure of 6·5 lb. per square inch at 1200 R.P.M. Deducting these, the piston friction is as shown in the curve, fig. 129, from which it will be observed that it falls from the equivalent of 9·5 lb. per square inch immediately after the fuel is cut off to 6·5 lb. per square inch after ten minutes, by which time it may be presumed that practically the whole of the carbonized oil on the cylinder walls has been replaced by fresh oil.

Influence of Temperature on Piston Friction.—As might be expected, the friction of the piston is largely dependent upon the temperature of the lubricant, and since the temperature of the latter is determined primarily by that of the cylinder walls to which it clings, it follows that the friction is controlled very largely by the temperature of the cooling water. In Chapter III, when dealing with the influence of cylinder temperature upon power and economy, it was shown that the indicated horse-power of an engine decreases with increase of temperature, because the reduction in the weight of charge far more than outweighs the slight gain due to reduced heat losses. In practice the brake-horse-power and economy of an engine generally increase with increase of temperature, because the reduced piston friction more than compensates for the reduction in indicated power. Fig. 130 shows the results of a test on a standard four-cylinder commercial vehicle engine when motored at a speed of 900 R.P.M. In this test, after the engine had been run for some time the water-jackets were thoroughly flushed through with cold water until the cylinders had been cooled down to atmospheric temperature. The supply of circulating water was then cut off and the rise in temperature and the friction torque (expressed in terms of mean pressure on the engine) were recorded at intervals of two minutes.

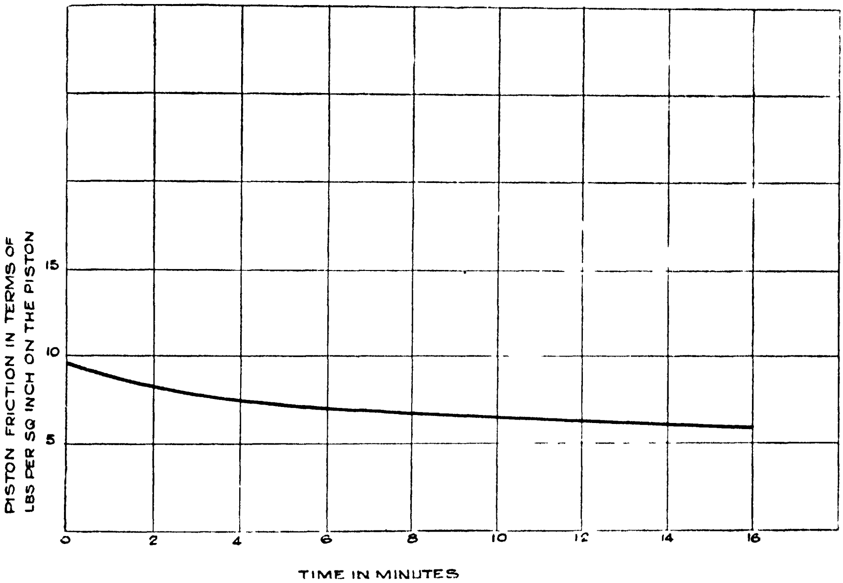


Fig 129.–Curve showing Drop in Piston Friction alone from fig. 117

In the case of this engine, the friction of the bearings and auxiliaries and the fluid pumping losses were determined separately, and were found to amount to the equivalent of 5·5 lb. per square inch at 120° F. If this figure be deducted throughout, it will be observed that the piston friction falls from 10·5 lb. per square inch with a cylinder temperature of 70° F. to 5·6 lb. per square inch with a temperature of 150° F. Over this range of temperature the decrease in indicated mean pressure, in the case of this particular engine, is about 2 per cent or 2 lb. per square inch, but the drop in piston friction is equivalent to about 5 lb. per square inch, so that the net increase in power at the higher temperature would be about 3 lb. per square inch or 3 per cent. This agreed fairly closely with actual results obtained when running under power, when the difference was found to be nearly 4 per cent. The conditions are not, however, exactly comparable, because

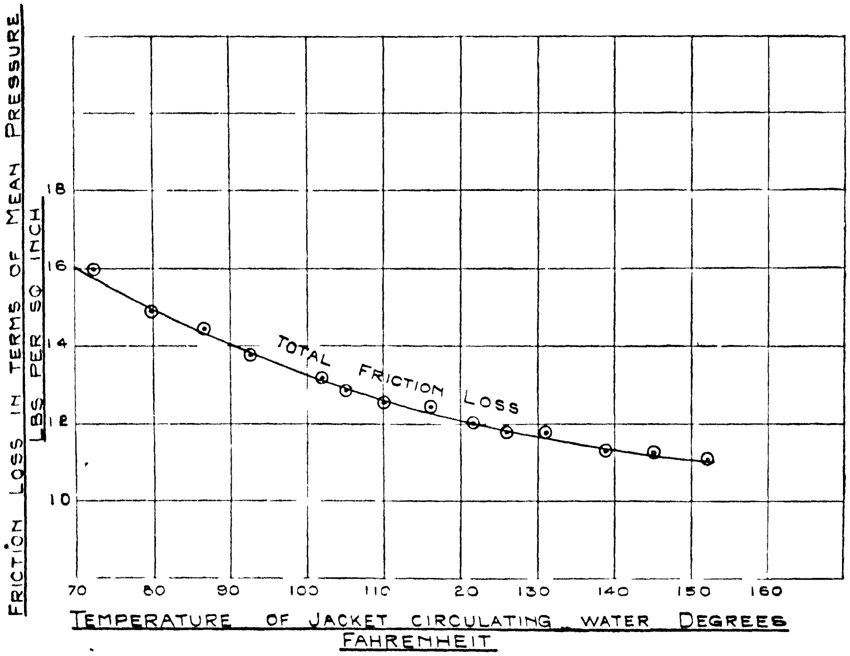


Fig 130 —Motoring Tests, showing Change of Friction with Temperature at 900 R P M

(1) When running under power the temperature of the piston and inner surface of the cylinder walls is higher than that of the jacket water; this would tend to reduce the difference in piston friction between hot and cold, because the whole temperature scale is virtually raised.

(2) When motoring, the oil on the cylinder walls was clean, consequently the piston friction was lower ; under actual running

conditions the piston friction was no doubt about 30 per cent greater at all temperatures; this would accentuate the difference, and probably more than outweighs (1).

Dissipation of Heat.—The heat from the crown of the piston is disposed of—

(1) Through the rings to the cylinder walls.

(2) Through the bearing surface to the cylinder walls.

(3) To the oil and air below the piston.

There is a great deal of evidence in support of the theory that, in comparatively small engines at all events, the bulk of the heat passes to the cylinder walls *via* the piston rings. The author could cite numerous experiments in confirmation of this theory, but it is probably sufficient to state that experiments have shown that, when all transference of heat by way of the bearing surfaces has been cut off, the temperature of the piston crown is found to be very little higher. In any case, it is evident that heat can only be transmitted rapidly to the cylinder walls through that portion of the skirt or bearing surface which is being pressed hard against them by the thrust of the connecting-rod and where the oil film is therefore both thinnest and in most active movement.

The proportion of heat carried away by the circulation of the air and oil below the piston cannot be very large and need not be taken seriously into consideration, except in cases where special arrangements are made to increase these effects.

It is quite clear that the most important consideration is the transmission of heat from the centre of the crown to the circumference; once the heat can be conveyed to the circumference, experience shows that there is no difficulty in getting rid of it. In order to facilitate the transmission from the centre to the circumference, it is obvious that the crown should be made as thick as possible consistent with the weight limitation, and the conductivity of the material should be as high as possible. During the last few years the use of aluminium alloys has come into vogue for pistons; not only is their weight about one-third that of cast iron, but their conductivity is about 2½ times as great. With such alloys, it is found that the rate of heat transmission is so high that, even in the case of cylinders developing over 120 B.H.P., there is no need to make the crown of the piston any thicker than is needed for strength alone. All aero-engines, and many others also, are fitted with pistons made throughout of aluminium alloys. The objections to an all-aluminium piston are:—

(1) Owing to the very high rate of expansion with temperature a large clearance must be allowed; this causes an audible knock when the thrust is reversed under pressure at the end of the compression stroke.

(2) Aluminium castings, unless very carefully annealed, are liable both to grow and to distort, so that yet more clearance must be allowed on this account.

Neither of these objections applies when the crown and ring-carrying portion alone is made of aluminium alloy and the bearing surfaces of cast iron, while advantage can still be taken of the high conductivity and light weight of aluminium.

Passage of Oil into the Combustion Chamber.—One common form of trouble with internal-combustion engines, and more particularly with those of the high-speed enclosed type, is the passage of oil into the combustion chamber, where it carbonizes both on the walls of the chamber and on the crown of the piston, and so gives rise to detonation and ultimately to pre-ignition. Passage of oil past the piston rings and so into the combustion chamber is due to the following:

(1) The oil is forced up against the rings on the downward stroke of the piston because the motion of the piston, combined with its thrust against the cylinder walls, sets up a very considerable hydraulic pressure and the oil is, so to speak, rolled up against the rings.

(2) The motion of the piston rings in their grooves tends to pump the oil into the combustion chamber.

In order, as far as possible, to prevent the passage of oil into the combustion chamber, the following considerations should be taken into account:—

(1) The setting up of a heavy hydraulic pressure can largely be prevented by perforating the bearing surface so that the pressure can relieve itself, and also by freely venting the piston just below the bottom ring.

(2) As the piston travels downwards the rings are all bearing against the top faces of the grooves, and the clearance between the lower sides of the rings and their grooves is filled with oil scraped from the cylinder walls. As the piston rises again, the rings change over and bear against the lower face of the groove; the oil therefore passes round behind the rings to their upper face and, at the top of the stroke, when the rings again change sides, some of it is squeezed out. It will be seen therefore that each ring functions

as a valveless oil pump and tends to deliver oil into the combustion chamber.

In order to reduce this pumping as far as possible—

(1) The rings, with the exception of the scraper ring, should be made as close a fit as possible in their grooves.

(2) Ample venting should be provided below the scraper ring to permit of the free escape of any oil scraped off the cylinder walls.

(3) The tendency of the rings to pump oil can further be checked to a large extent by drilling holes through the ring groove behind the ring, thus permitting of the free escape of any oil as it passes the back of the ring.

This latter expedient should apply only in the case of the lowest ring, since such drilling permits also of the escape of gas. Fig. 131 shows an arrangement of rings, &c., which has been found very effective in preventing the passage of oil into the combustion space.

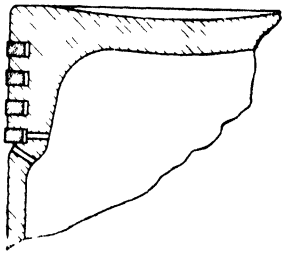


Fig. 131.—Section of Slipper Piston, with Scraper Ring vented by Holes drilled through Groove

There is a widespread belief that the passage of oil into the combustion chamber is dependent primarily upon the pressure or vacuum in the cylinder, and that, when throttled down, the reduced pressure in the cylinder during the suction stroke causes oil to be sucked up past the rings. This belief is founded upon the fact that when an engine is opened out after running throttled up, smoke becomes apparent, indicating an excess of oil in the combustion chamber. Also when an engine has been run with the throttle nearly closed and the cylinders or valve caps are removed, oil in a liquid state is then found in the combustion chamber. In spite of such evidence, however, the theory is quite fallacious, for the actual quantity of oil passing the rings is found to be a function of the speed, and of the speed alone, the pumping pressure set up by the rings being of far too high an order to be influenced appreciably by any relatively slight differences of pressure in the cylinder. When an engine is running at or near its full load, the oil passing into the combustion chamber is burnt along with the fuel; combustion is nearly complete, so that no smoke is visible from the exhaust. When running dead light the flame temperature, owing to the large dilution with exhaust gases and the relatively higher rate of heat loss, is insufficient to burn the oil, with the result that it accumulates in the combustion chamber until the throttle is opened; the quantity is then so large that there is not sufficient oxygen available for its complete combustion at first, with the result that it is only partially burnt and issues from the exhaust as blue smoke—that is, as partially vaporized but unburnt oil.

In any normal high-speed closed-type engine, about 90 per cent of the lubricating oil consumed is burnt in the cylinder as fuel, a fact which should always be borne in mind when reckoning the efficiency of an engine, for the hourly consumption both of fuel and oil should be taken into account. In most normal engines the proportion of oil consumed is very small in relation to the fuel and does not materially affect the consumption of the latter, but in the case of certain aero-engines, particularly of the rotating cylinder type, the consumption of oil is so high as materially to reduce the fuel consumption, and so give rise to a fictitious fuel economy.

The author has carried out a number of tests in order to ascertain the influence of both pressure and speed on the passage of oil into the combustion chamber, and has tried the effect of motoring an engine and collecting the oil passing the piston under the following conditions:

(1) When the pressure on either side of the piston is atmospheric.

(2) With a continuous vacuum of 20 in. of mercury in the cylinder.

(3) With a continuous air pressure of 45 lb. per square inch above the piston.

In all three cases the quantity of oil passed per hour was, within the limits of observation, the same certainly to within 10 per cent. In all cases also the quantity of oil passed by the piston varied almost directly as the speed of rotation.

In fig 132 is shown a special machine used for testing pistons and rings for -

(1) Friction.

(2) Leakage.

(3) Passage of oil.

It consists of a water-jacketed cylinder mounted on a crankcase in which different types of pistons, rings, &c., can be fitted. The piston is reciprocated by means of a crank and connecting-rod, and, to ensure freedom from vibration, reciprocating balance weights, operated by eccentrics, are provided. The cylinder is heat-insulated from the crankcase and the friction of the piston is measured directly by the temperature rise of the water in the jacket. In order to ensure uniformity of temperature the water in the jacket is kept circulating by means of a small propeller driven by a belt from the crankshaft. The top end of the cylinder is connected to a large and heavily lagged receiver of sufficient capacity to prevent any

Fig. 132.—Photo of Machine designed for testing Pistons

appreciable variation in pressure, so that the same air is drawn in and out of the cylinder at every stroke, and errors, due to the circulation of cold air inside the cylinder, are eliminated, or nearly so. This receiver is, in turn, connected to an air pump so that the pressure on the piston can be raised or lowered to any desired degree, and the effect of fluid pressure both on piston friction and the passage of oil can be observed.

For lubrication, oil is forced under a pressure of 30 lb. per square inch through the hollow crankshaft, from which it passes out through the connecting-rod big-end bearing and is thrown on to the cylinder walls.

For testing gas tightness and leakage of rings the receiver is removed and a plain cover fitted in its place on the top of the cylinder. This cover is provided with a small and very light automatic inlet valve connected to an air-measuring device. With this cover fitted the piston alternately compresses and expands the air in the cylinder, the maximum pressure being about 220 lb. per square inch. Any leakage past the rings is made up by air entering through the inlet valve, and the quantity of air so required to make up for leakage is measured by the displacement of water.

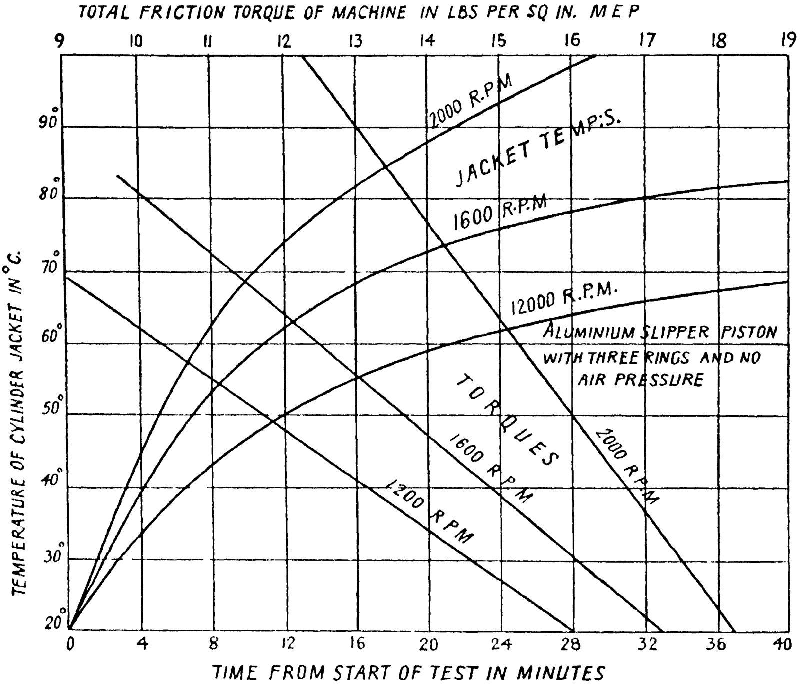


Fig 133 —Test Readings at three different Speeds

The machine is direct-connected to a balanced swinging field electric dynamometer, and can be driven at any speed from 600 to 2500 R.P.M. The total torque required to rotate the machine at any speed can be measured direct from the arm of the dynamometer, while the piston friction alone can be determined directly from the temperature rise of the water in the cylinder jacket.

Owing to losses by radiation, &c., it is not easy to determine accurately the absolute friction of the piston, but the relative friction as between two pistons, or any variations in piston design or rings, can be measured with extreme nicety by comparing the curves of temperature rise of the jacket water. Figs. 133, 134, and 135 show a number of such curves of temperature and time and also the total friction of the machine as a whole in terms of lb. per square inch on the piston, with various types of piston, numbers of rings, &c.

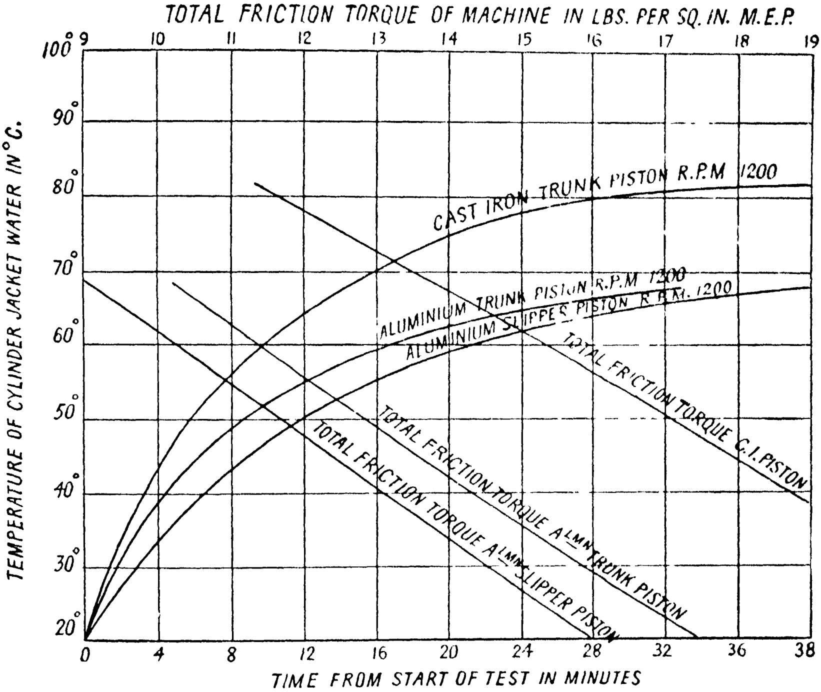


Fig. 134.—Test Readings with three different Pistons. Speed 1200 R.P.M. in all cases

In order both to reduce weight and to prevent distortion, it is clearly very desirable to transmit the pressure as directly as possible from the crown of the piston to the connecting-rod, and from the connecting-rod to the bearing surfaces. The customary method of transmitting the pressure from the crown through the side walls and ring grooves to the two extreme ends of the gudgeon pin has nothing to recommend it. It is clearly far better to transmit it directly from the crown of the piston to the gudgeon pin at points as near the centre as the width of the connecting-rod small end bearing will permit.

Fig. 136 shows a design of all-aluminium piston, and represents probably the lightest possible construction. In this design two main ribs transmit the load from the crown to the gudgeon pin, and from the gudgeon pin to the bearing surfaces. Also all unnecessary bearing surface has been eliminated. This type of piston has come to be known as the slipper type, and has been widely used, particularly in very high-speed engines, where its light weight, low friction loss, and effective oil-resisting properties have rendered it of great advantage. The example shown in fig. 136 is the largest yet made, for this single piston transmits 135 B.H.P. at 1400 R.P.M.

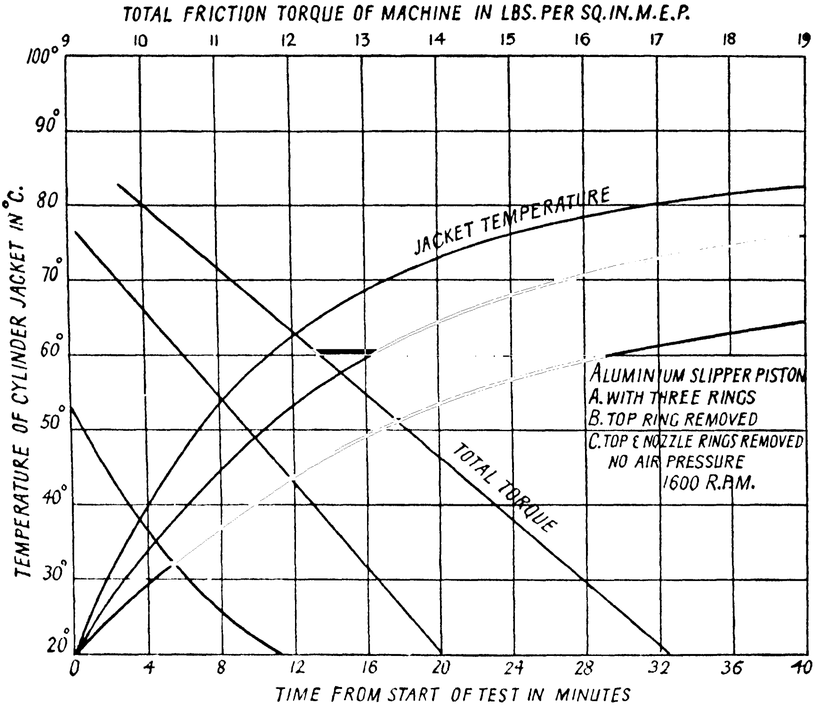


Fig. 135.—Test Readings with different Numbers of Piston Rings. Speed 1600 R.P.M. in all cases

Fig. 137 shows an alternative design in which the slipper or bearing surfaces are of cast iron. This design has the further advantage in that the floating gudgeon pin is located by the sides of the cast-iron sleeve and requires no other means of endwise location—always rather a troublesome problem.

Fig. 138 shows yet another design, in which the aluminium alloy head is connected with the cast-iron cross-head portion by means of the gudgeon pin, or rather by means of the bushes in which the gudgeon pin floats. This design has the advantage that it is cheaper and less fragile than that shown in fig. 137, also that the aluminium head is free to centre itself in the cylinder. The chief objection to it is that it necessitates very accurate workmanship.

Piston Knock.—Owing to the large clearance which must be allowed when all-aluminium pistons are used, it is very difficult to obtain silent running, for at the end of the compression-stroke the piston is thrust violently from one side of the cylinder to the other. The noise is most apparent when an engine is running slowly on a light load; under such conditions the piston is cool, and therefore the clearance is at a maximum, also the other mechanical noises of the engine are less apparent. Various devices have been experimented with in the endeavour to overcome this troublesome noise. Some designers have used pistons in which the normal clearance is very small and the skirts are slit in order to allow some elasticity

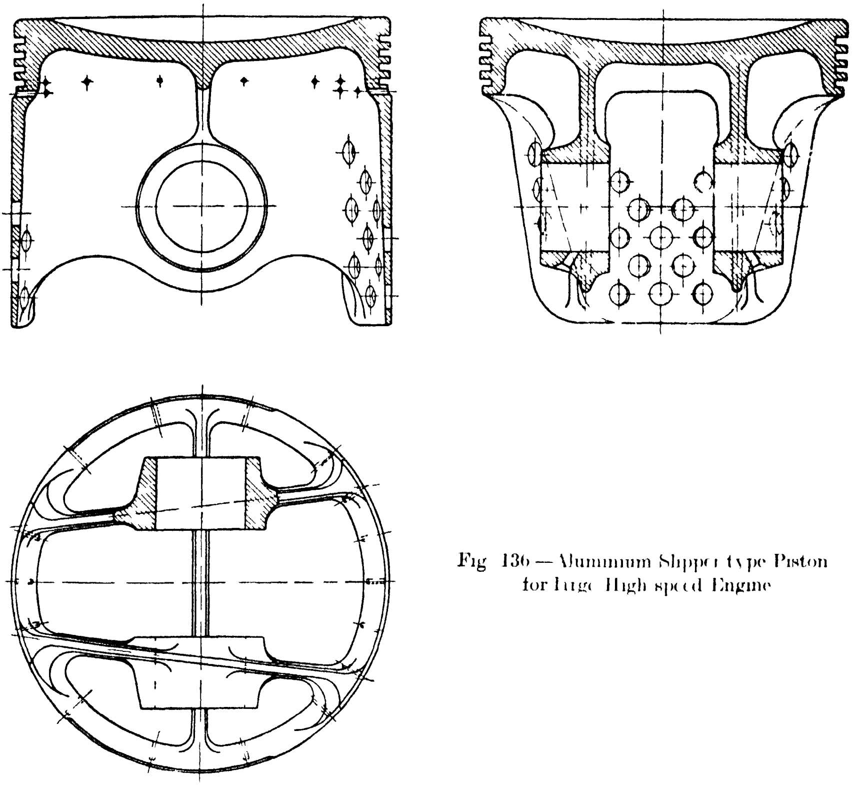


Fig. 136.—Aluminium Slipper type Piston for large High speed Engine

and prevent seizures; others, again, have even gone so far as to introduce springs between the connecting-rod and piston, in order to keep the latter bearing at all times against one wall of the cylinder. Others, again, have adopted the method of fitting the gudgeon pin out of centre in the piston so that the latter tends to tilt about the gudgeon-pin centre. This latter method appeared promising at first sight, but on further investigation it was found to be ineffective. The precise effect of offsetting the gudgeon pin in this manner is

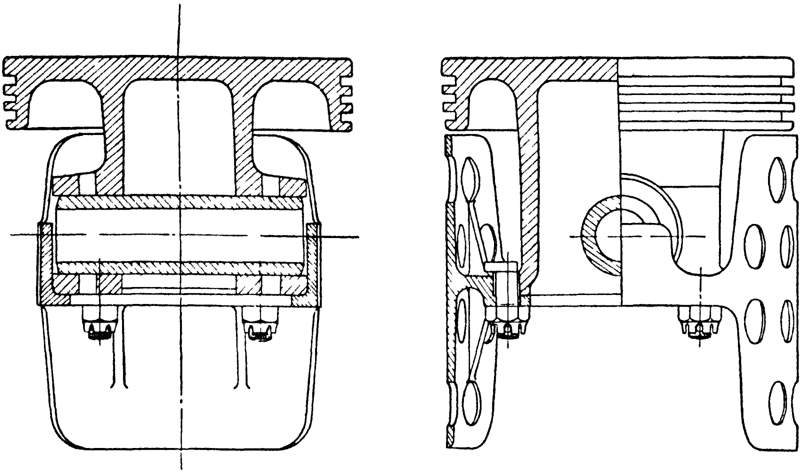


Fig. 137.—Composite Piston with Aluminium Head and Cast iron Bearing Surfaces rigidly attached

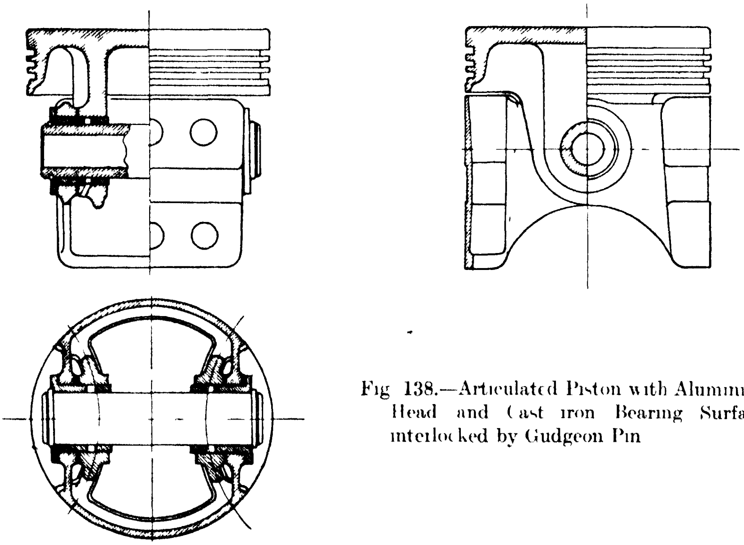


Fig 138.—Articulated Piston with Aluminium Head and Cast iron Bearing Surfaces interlocked by Gudgeon Pin

illustrated in fig. 139, which shows the position taken up by the piston at various points in the cycle.

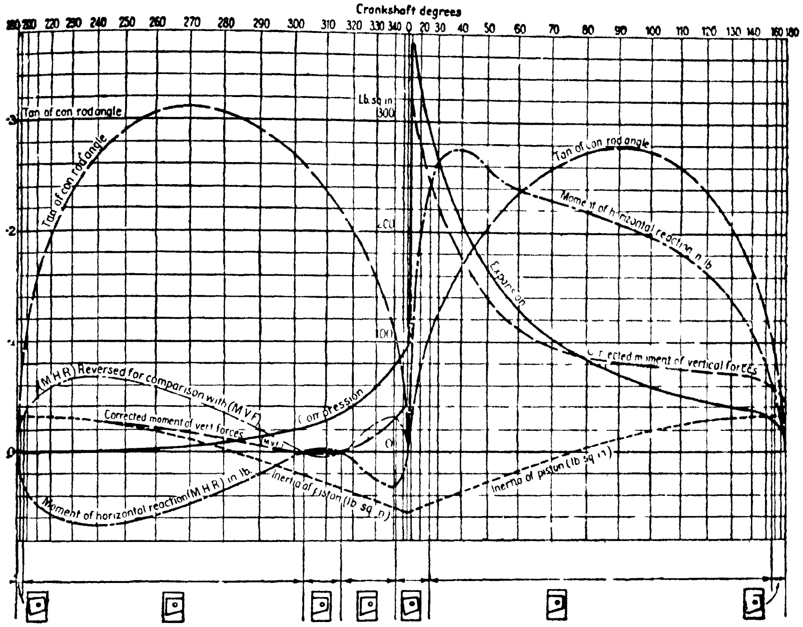


Fig. 139.—Diagram of Forces on Piston with offset Gudgeon Pin, showing effect on clearance

Fig. 140 shows a method patented by the author and applied to a slipper-type piston in which the bearing surface is severed from the crown in order to prevent the direct transmission of heat and so permit of a smaller clearance being used. This arrangement proved very successful in reducing the passage of oil past the piston rings because of the exceptionally free venting of the oil below them, but it did not permit of any appreciable reduction in clearance, for the simple reason that very little heat is transmitted from the crown to the bearing surfaces in any case; in other words, it was found that, at all events in the case of comparatively small engines, the temperature of what may be termed the cross-head portion of the piston was in any case very little in excess of that of the cylinder walls, so that insulating it from the crown had little influence on its temperature, and therefore on its expansion.

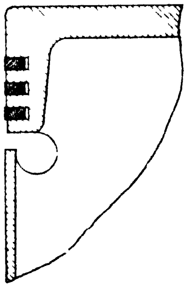


Fig. 140.—Split Slipper Piston

None of the methods described above can be said to have solved the difficulty of piston knock, or even to have gone very far towards solving it, and the author is inclined to the opinion that where extreme silence is required it is better to employ cast iron for the bearing surfaces in all except very small pistons, in which the clearance may be so small that there is little or no knock in any case, for the noise is dependent upon the absolute, rather than upon the proportionate, clearance. Experience has shown that with all-aluminium pistons fitted to a water-cooled engine of average performance it is necessary to allow a clearance on the bearing surfaces of approximately 0·002 in. per inch of diameter. As a general statement, when the total clearance exceeds from 0·005 in. to 0·006 in. piston knock becomes audible, that is to say, all-aluminium pistons up to 3 in. can be made to run silently, depending upon the lubrication and a number of other minor controlling factors; but above 3 in. diameter it is extremely difficult, if not impossible, to ensure silent running.

Piston Rings.—Generally speaking, piston rings do not call for much comment. With but few exceptions all high-speed internal-combustion engines employ ordinary plain concentric cast-iron rings of the Ramsbottom type. Such rings should always be ground both on the face and sides in order to ensure a close fit in the grooves, and should preferably be hammered, after being split, in order so to stress the material as to ensure a uniform pressure against the cylinder walls. The most important feature to ensure is that they fit truly against the sides of the grooves, since this determines largely the amount of oil they will pump. The top gas ring should be allowed a few thousandths clearance in order to keep it from sticking and, when the scraper ring groove is perforated, this ring also should be allowed liberal side clearance. When aluminium pistons are used there is always a tendency for the edges of the " lands " in the piston to be dragged over by particles of grit and so to lock the rings in their grooves. This tendency can, however, be prevented by chamfering slightly the edges of the grooves. This is a small point, but it is one which should not be overlooked, for its neglect has probably done more than anything to prejudice unjustly the use of aluminium for pistons.

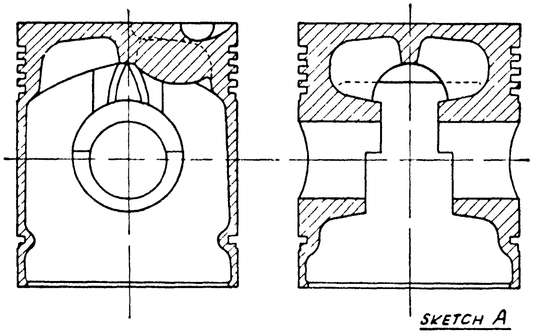
Width of Ring.—All reasoning points to the conclusion that piston rings should be made as narrow as possible so long as they are not too fragile to machine or handle. For a given radial thickness, the narrower the ring the less both the friction and the inertia hence the lower the total pressure against and therefore the wear on the sides of the ring grooves. There, is, however, some evidence in support of the theory that an excessively narrow ring causes increased wear of the cylinder bore due to the comparative ease with which the oil film breaks down during the period the piston is at rest on top dead-centre.

Radial Thickness.—The radial thickness determines the spring tension against the cylinder walls. So long as this is above a certain figure there is no object in increasing the thickness, which merely involves an increase in friction and wear on the cylinder walls.

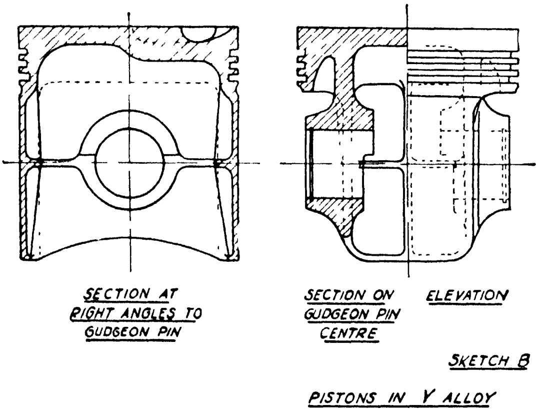
There is definite evidence to indicate that, when working, the rings are pressed out against the cylinder walls by the gas pressure behind them, and that the pressure in the ring groove is on the average something less than the mean pressure of the cycle. In fact recent experiments by the Admiralty Engineering Laboratory, in which indicator diagrams were actually taken from behind the piston rings, showed that the pressure behind the top ring followed very closely that of the gas pressure above the piston, while that behind the second ring showed a very much delayed diagram, and behind the third and others a more or less uniform pressure scaling down through the remaining pack of rings. Experience indicates that in a well-made ring a spring pressure of from 6 to 7 lb. per square inch is sufficient, and that any further pressure results merely in extra friction without any compensating advantage. The spring tension required, however, depends to some extent upon the amount of clearance between the "lands" of the piston and the cylinder walls. It is customary and proper to make the clearance of the "lands" such that they will not, under any circumstances, touch the walls of the cylinders. If, however, the clearance is too great, a considerable area of the side of the ring may be exposed to the full fluid pressure, and, as a result, the ring may be pressed so hard against the lower face of the groove that its spring tension will not overcome the friction against the side of the groove; under these conditions the ring will become locked and unable to expand against the cylinder walls, with the result that severe "blowing" will occur.

There is also very strong evidence to show that the top land should always be as deep as possible, in order to provide adequate protection for the top piston ring. In the designs of piston illustrated previously, the top land is in most instances rather too shallow, due to the fact that the available height from the gudgeon-pin centre to the piston crown was, in each case, so restricted as *to render the provision of an adequate protecting land impossible.*

Pistons for High-duty Diesel Engines.—In the above remarks on the open or trunk type piston much of what has been said applies equally to the petrol or Diesel engine. Generally speaking, however, the piston constructions illustrated are more suitable for



Standard Piston
Weight, including rings, gudgeon pin and circlips, 5·34 lb. (2·43 kilograms)


SECTION AT RIGHT ANGLES TO GUDGEON PIN
SECTION ON GUDGEON PIN CENTRE
ELEVATION
SKETCH B
PISTONS IN Y ALLOY

Slipper Piston
Weight, including rings, gudgeon pin and circlips, 4·16 lb. (1·89 kilograms)

Fig. 141

petrol than for Diesel operation, where the higher maximum cylinder pressures call for a rather more robust design. Fig. 141 gives two pistons designed for a high-speed Diesel engine of $4\frac{3}{4}$ in. bore, of speed range up to 2400 R.P.M. Sketch A is an illustration of a typical trunk construction, in Y alloy, fitted with three gas rings and two scrapers. It will be seen that the piston conforms with the general principles given in the earlier part of the chapter, and is provided with vent holes and relief below the scrapers to limit the flow of oil into the combustion chamber. Sketch B shows a lighter design for the same engine. of slipper type, similar to those previously illustrated but of rather more robust construction. By thus relieving the oil pressure on the rings, it was possible to do away with one pressure ring and one scraper without increasing the lubricating oil consumption and with a marked improvement in performance due to reduction in running friction. The curves of recent comparative tests on the two pistons given in fig. 142 illustrate this point.

Cross-head Piston.—All the foregoing remarks refer to the open or trunk type of piston, and more particularly to that form of open piston which has come to be known as the slipper type, a form with which the author has had the most experience.

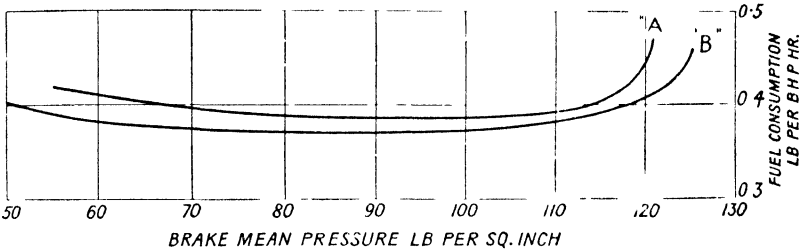


Fig. 142.—Comparative Fuel Consumption Curves with Trunk and Slipper Type Pistons in the same Engine

Curve A—Trunk type Y-alloy piston. Curve B—Slipper type Y-alloy piston.

In the larger sizes of high-speed engine more particularly, the author prefers to use wherever possible a somewhat different type which is now generally known as the cross-head piston.

In this design the two functions of an open piston—namely, to act as a piston proper and as a cross-head guide—have been separated to a far greater extent than in the case of the slipper type, until it resembles much more nearly the usual steam-engine form.

The piston itself consists of an ordinary flat or concave crown carrying the piston rings and a plain light tubular stem extending from the crown of the piston to below the gudgeon pin. The lower portion of this trunk is surrounded by a steel or cast-iron sleeve, which embraces and locates the floating gudgeon pin, and constitutes the only wearing surface. This cross-head sleeve runs in a cylindrical guide, which is spigoted both into the cylinder and the crank chamber, or in some cases into the cylinder only, which is prolonged to accommodate it. A general arrangement of the piston, cylinder, and cross-head guide is shown in fig. 143, from which it will be observed that the crown of the piston serves only to carry the rings and transmit the pressure down the hollow cylindrical stem to the gudgeon pin. It does not bear upon the cylinder walls at all, and therefore requires only just sufficient lubricant to maintain the rings in good condition. It will also be observed that in this construction the cylinder walls are cut off from all splash lubrication.

In spite of the fact that this piston is some 30 per cent heavier than the slipper type, the total friction is little more than 80 per cent that of the slipper piston and only about 60 per cent that of a piston of the ordinary trunk type.

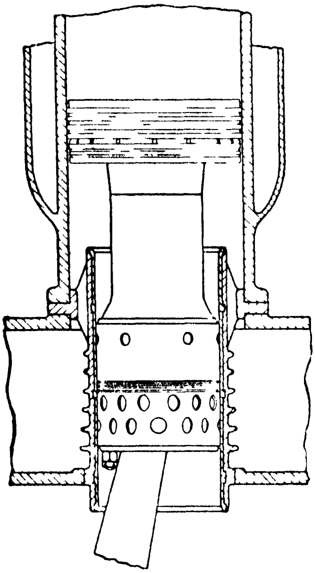


Fig 143 —Part Section showing Cross-head Piston and Guide

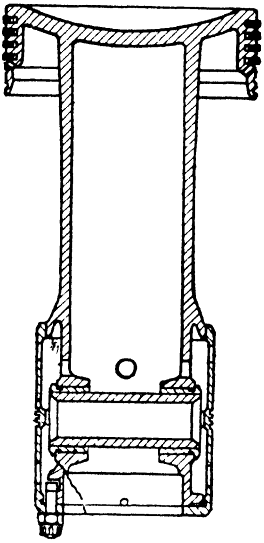


Fig. 144 —Section of Piston

The construction of the piston is shown in detail in fig. 144, and the photographs reproduced in figs. 145 and 146. The cross-head sleeve is an easy push fit over the lower portion of the stem, and is held in place by four small bolts; it is supported by three bearing lands, one at the middle on the gudgeon-pin centre line and one at each end.

In ordinary commercial engines this sleeve is made of cast iron, but in very high-speed engines a light high carbon steel sleeve is used. When employing this piston on engines having separate cylinders the guide is spigoted into both the cylinder and crankcase, and the latter is provided with a false top as shown in fig. 143 above.

The space between the false top and the base of the cylinder is utilized for the circulation of air round the cross-head guide, which is ribbed for cooling. The air enters at one side and passes out to the carburettor on the other side. A portion of the air is drawn directly round the cross-head guide, and the remaining portion passes between the guide and the cylinder through slots provided for this purpose. On the upward stroke of the piston the air is drawn through these slots at a high velocity, and impinges against the crown and stem of the piston, thus effectually cooling them. On the downward stroke this heated air is discharged again into the chamber surrounding the guides, and thence into the carburettor. By this means the piston and cross-head guide are kept cool and the carburettor air is warmed.

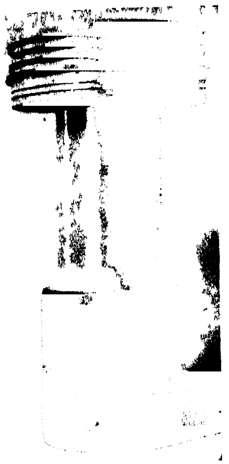


Fig 145 —Cross head Piston

It is found in practice that the heat abstracted from the piston

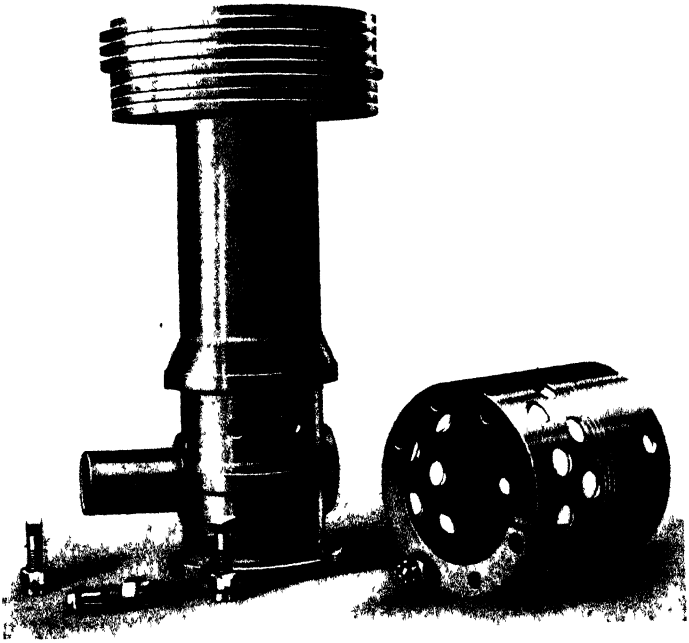


Fig. 146.—Cross-head Piston dismantled

and cross-head guide is just sufficient for good distribution with petrol of high boiling-point and low volatility. Tests carried out on several of the engines built for tanks with thermometers fitted in the induction piping above and below the carburettor have shown that when running on full load with an atmospheric temperature of 60° F. the air, after passing round the cross-head guides and pistons, entered the carburettor at a temperature of 130° F.

On a light load, with consequent reduced air circulation, the temperature of the air entering the carburettor rose to 150° F., and the temperature near the top of the induction pipe to 100° F., which is sufficiently high to check condensation at reduced loads. The free circulation of air through the upper portion of the crankcase tends to keep the lower portion cool, so that no oil cooling is required.

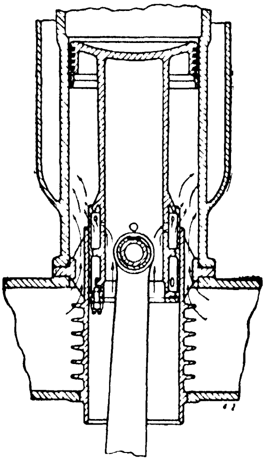


Fig 147.—Cylinder Lubrication with Cross-head Piston

The system of cylinder lubrication is shown in fig. 147. The lower portion of the stem of the piston is provided with a few small holes, and the cross-head sleeve which surrounds it is also provided with a ring of small holes so placed that these holes are uncovered above the guide at the top of each stroke. On the upward strokė of the piston, air is drawn through slots provided in the flange of the cross-head guide between the guide and the cylinder, and passes at a high velocity around the cross-head sleeve; in doing so, it draws a small proportion of air and oil mist from the holes in the cross-head sleeve, which are in communication with the crank chamber through corresponding holes drilled in the piston stem. The oil issuing from these holes in the form of a mist is picked up by the rush of air and sprayed over the cylinder walls while the piston is near the top of its stroke; the total quantity of oil drawn out in this manner is minute, but it is sufficient for the maintenance of the piston rings. The whole operation is similar to that of a spray carburettor in which the slots in the cross-head guide correspond to the choke tube, and the holes in the sleeve to the jets The control of the quantity of oil delivered in this manner is governed by the area of the slots and the size or number of holes provided in the sleeve.

It will be seen that, by this means, the lubrication of the cylinder walls is continuous, that oil is only supplied to the cylinder walls in the quantity required by the piston rings, and that oil which has clung to the walls and become partially carbonized does not find its way back into the crankcase. The provision in this manner of an entirely separate system of lubrication to the cylinder allows of the use of unstinted lubrication to all the other working parts without the risk of carbonization of the piston or any tendency to smoke; also, the oil consumption is exceedingly low.

When working with kerosene or high boiling petrols, this type of piston is particularly suitable, for one of the chief troubles with such fuels is that they tend to precipitate upon the relatively cool walls of the cylinder barrel and so to pass down into the crankcase, thus contaminating the lubricating oil, and cause trouble with the bearings. With the cross-head type piston, however, any fuel which may succeed in passing the piston is trapped in the chamber surrounding the cross-head guides, from which it may be drained off before it can do any harm. The quantity of kerosene which, in practice, is drained away from this chamber is often surprisingly great, particularly when working on variable loads, often amounting to as much as from 4 to 8 per cent of the total fuel consumption of the engine, or from three to six times the oil consumption.

The advantages of this type of piston may be summarized as follows:—

(1) The lubrication is under complete control, and is independent of the crankcase lubrication; consequently the oil consumption, the tendency to carbonize both the piston and combustion chamber, and the risk of oiling up the sparking plugs are all reduced to the minimum.

(2) The piston friction is reduced to little more than half that which obtains with an ordinary trunk piston.

(3) Owing to the fact that the cross-head and guide are relatively

cool, and that both are maintained at approximately the same temperature, a very fine running clearance can safely be used, thus ensuring silent running.

(4) Since the piston itself does not bear upon the cylinder walls, an ample working clearance can be allowed without any risk of noise.

(5) The wear on the cylinder walls is reduced to a minimum, since only the piston rings bear against them and there is no side thrust.

(6) The gudgeon-pin being short, stiff, and free to rotate, and also being placed in such a position that it receives very little heat from the piston, does not wear perceptibly.

(7) The bulk of the heat from the crown of the piston and from the cross-head guide is utilized to warm the air for the carburettor, and is not transferred to the crankcase.

(8) All the working parts can be lubricated without stint and without any risk of excess of oil reaching the cylinder walls; also, the oil remains clean.

(9) In the event of any fuel condensing on the walls of the cylinder, its subsequent passage into the crankcase can be prevented absolutely.

(10) The restricted lubrication to the cylinder walls greatly reduces any tendency of the piston rings to become carbonized or gummed up.

(11) There is no tendency for the engine to become "gummed up" when cold.

The principal objections to the use of this type of piston are:—

(1) That it increases the height of an engine as compared with the use of an open-type piston, by an amount equal to about two-thirds of the piston's stroke.

(2) That it necessitates an engine designed specifically for its use, and unless separate cylinders are used it introduces difficulties in the way of alignment between the cross-head guides and the cylinder bore. These difficulties are not, however, insuperable, as is illustrated by the engine shown in fig. 148, which shows the application of cross-head pistons in the case of the engines manufactured by Messrs. Peter Brotherhood, Ltd., for tractors and marine work, in which the four cylinders and the upper half of the crankcase are cast in one piece.

(3) That it increases slightly both the cost and weight of an engine.

These objections are, in the author's opinion, easily outweighed

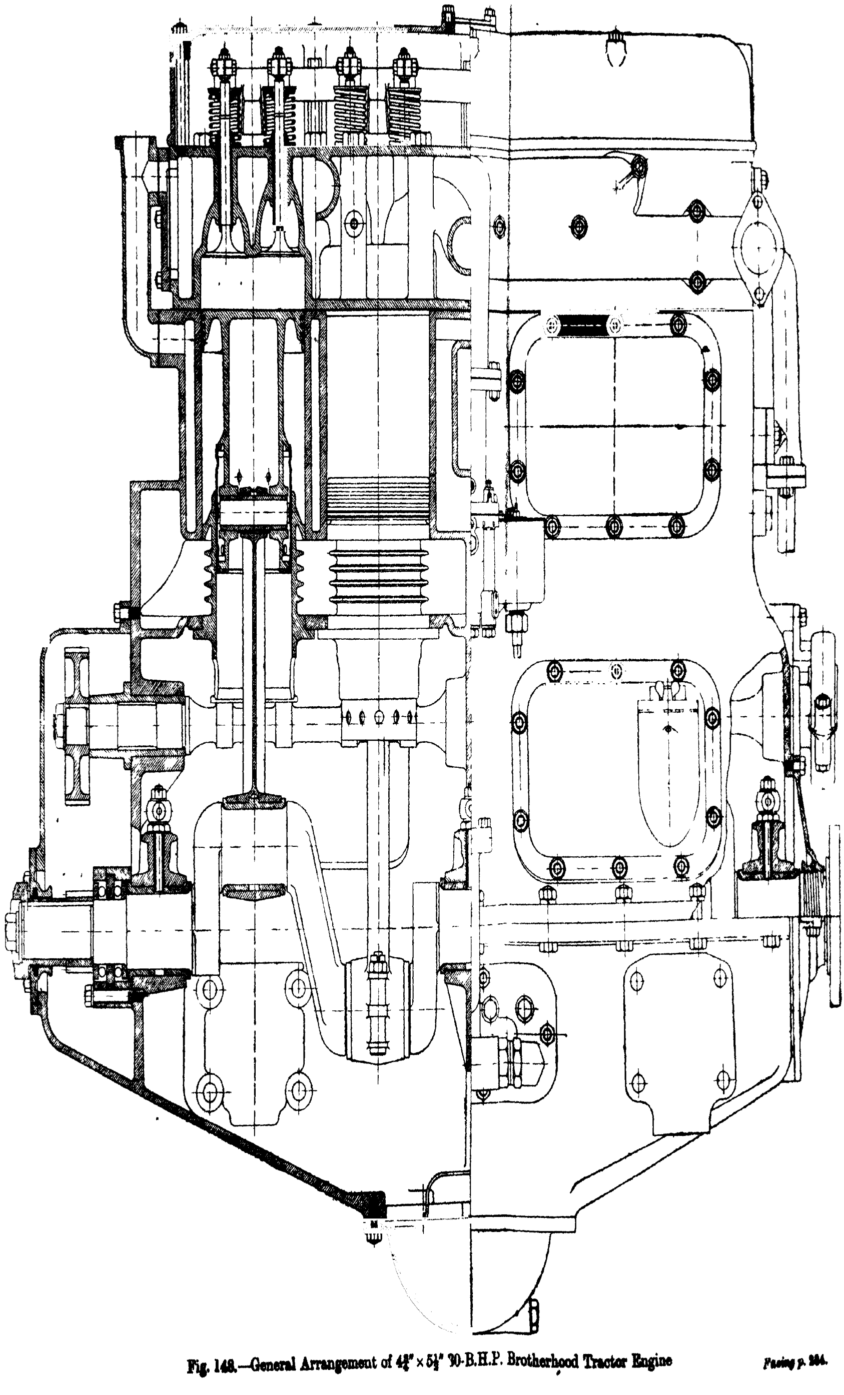


Fig. 148.—General Arrangement of 4¾″ × 5½″ 30-B.H.P. Brotherhood Tractor Engine

Facing p. 284.

by the advantages to be gained, more especially in the case of large engines, such as that in fig. 149 (p. 286), or in the case of engines using kerosene or petrol with a high final boiling point. In any case their use makes for a higher mechanical efficiency, silent running, and reduced cylinder wear and oil consumption.

Cylinder Wear.—The phenomenon of cylinder wear has of recent years come to the forefront as one of the major problems of engine design. The reason for this was the realization that wear is not solely due to physical abrasion, but is equally a chemical process. The way was thus opened up for a new line of research, involving investigations into the various factors in engine design and operation which, on the hypothesis that chemical action is the cause, might affect the rate of cylinder wear.

In 1932 the author put forward a suggestion, based on observations spread over a number of years on engines of different types and covering a wide range of operating speeds, that corrosion plays a large part in the process, and is indeed the predominating factor. Since that time investigations over a wide field have in no way disproved this theory.

The belief that corrosion is the main cause of cylinder wear was set down in the following terms[1]:

"As the piston approaches top dead centre, the protecting oil film between the rings and the liner begins to break down, and, at the same time, the pressure behind the ring begins to rise. At top dead centre the ring is stationary for an instant and the pressure behind it is rising rapidly; what little oil remains is squeezed out until no more than a layer of molecular thickness is left, and the surface of the liner at that point 'squeegeed' until it is practically dry. As the piston begins to descend, the pressure behind the ring is still building up, and some appreciable distance is travelled before the velocity is such that the ring will again be lifted and ride on a fluid film. Now as the piston descends further, there is left exposed to the flame a band of the liner of indeterminate depth from which the oil film has been wiped almost clean, and which is therefore largely unprotected from direct attack by the products of partial combustion. On this exposed sore they settle like flies and satiate their appetites with the bare metal. On the next cycle the surface is again scraped clean by the rings and prepared for a fresh attack."

Although much has been done to give the engine greater

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[1] "Notes and Observations on Petrol and Diesel Engines," H. R. Ricardo, *Proc. Inst. A.E.*, Vol. xxvii.

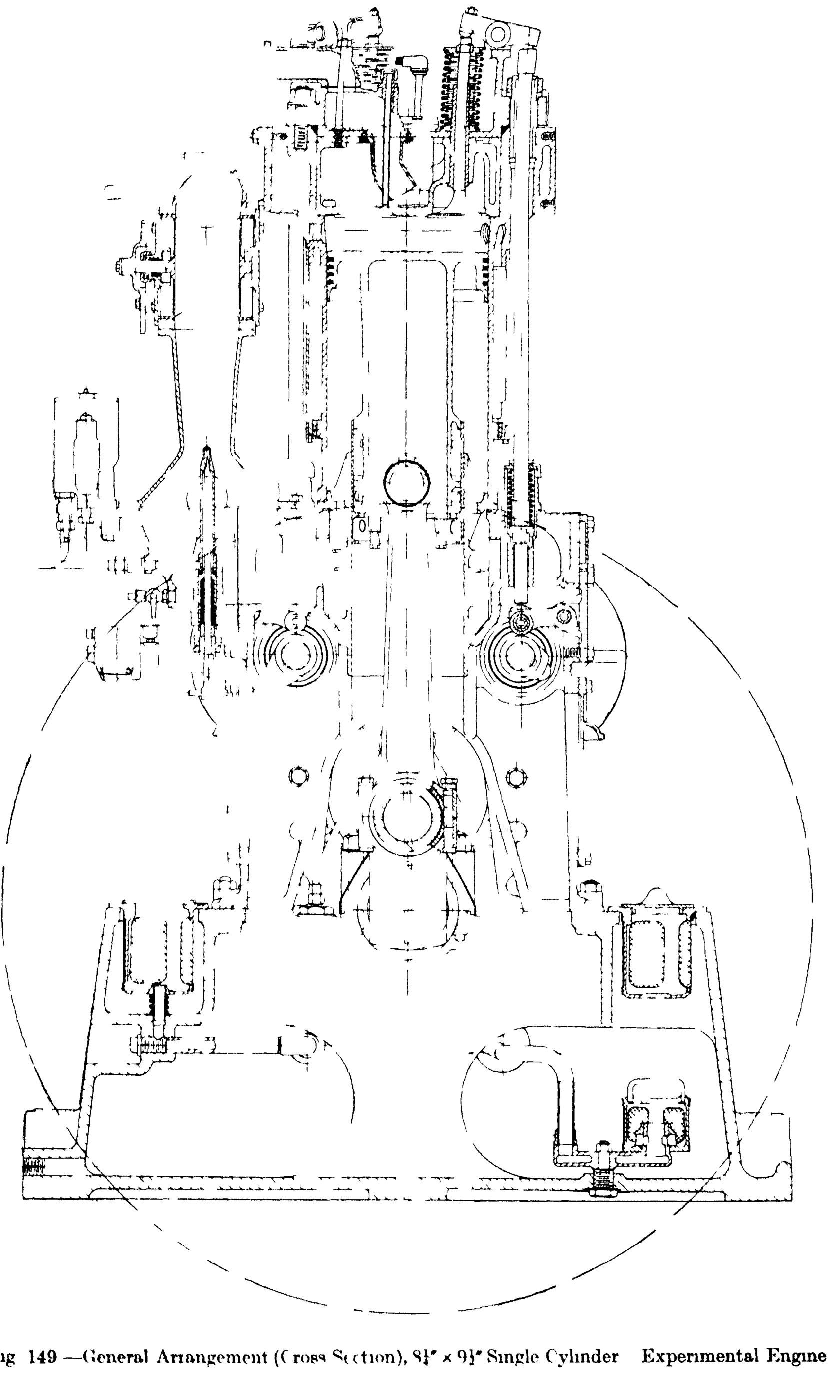


Fig 149 —General Arrangement (Cross Section), 8¼" × 9½" Single Cylinder Experimental Engine

mechanical reliability and to prolong the life of working parts, the wear of cylinder liners is still one of the bugbears of the internal-combustion engine.

In all cases the incidence of wear in relation to the piston stroke is the same. The region of greatest wear extends over the narrow band occupied by the ring assembly at T.D.C. The maximum wear over this band occurs at the limit of the travel of the top ring, and the variation in wear is such that at a level opposite the lowest pressure ring, it is very approximately one-sixth the value at the top ring level. For the remainder of the bore swept by the piston, the cylinder wear is negligible, except in severe cases. Occasionally there is a small amount of measurable wear opposite the ring assembly at B.D.C. Over this region the greatest wear is again opposite the top ring, but the relative variation is much smaller than at the T.D.C. position.

The following notes give briefly some of the conclusions drawn from many observations on different engines.

1. The average rate of wear at the upper limit of ring travel where the greatest wear occurs is of the order of 0·002–0·004 in. per thousand hours for petrol engines, and about 50 per cent higher for Diesel engines.

2. The surface hardness of the liner material appears to have but a small effect on the rate of wear. On the other hand, the resistance of the material to corrosion has a marked influence. The rate of liner wear is apparently unaffected by the material used for the piston. At one time it was thought that the use of aluminium alloy for pistons was responsible for an increase in liner wear. Subsequent experience with aluminium pistons disproved this.

3. It has been shown from many separate tests that there is no change in the rate of wear over a considerable range of lubricating oil consumption. If, however, a supply of oil far in excess of normal requirements is delivered to the cylinder bore, leading to an increased oil flow past the piston and a heavy oil consumption, some reduction in liner wear can be achieved

4. The rate of wear appears to bear no relation to the speed of operation, so that whatever the engine speed, the rate per thousand hours remains the same. This strongly suggests that it is the actual time of exposure of the surface to combustion products rather than the number of times the ring traverses the liner surface that is the important factor.

5. It is a well-known fact that the wear of the face of the ring

which bears on the liner is negligible compared with that of the liner itself. In the vicinity of T.D.C., where the greatest liner wear occurs, the oil is squeezed out by the pressure behind the ring acting at a time when the piston velocity is almost nil. Here we have a clear indication that it is corrosion of that part of the liner which is thus deprived of lubricant that causes wear to take place. The ring surface, which is always protected, does not suffer. In point of fact, the sides of the ring undergo a much greater degree of wear than the periphery. The wear in this case is caused by abrasion due to the efforts of the ring to accommodate itself both to the liner surface and to the tilt of the piston.

6. Another interesting observation is the relative immunity of the single sleeve-valve from wear of the sleeve bore. In this case we have a mechanism in which there is no cessation of motion between the piston and sleeve or moving liner. The conditions for boundary lubrication which occur in a normal reciprocating piston and stationary liner assembly consequently do not exist. It was found many years ago that wear in the single sleeve-valve engine was quite different in two respects: (*a*) it was not localized, and (*b*) it was only a fraction of the amount usual in the normal reciprocating engine, of the order of one tenth as much.

Subsequent research by the Institution of Automobile Engineers and by the Laboratory of the Royal Dutch Shell Co. at Delft has confirmed that it is the action of products of partial combustion that is responsible for the greater part of the wear. These laboratories have done much really brilliant and valuable work in separating the various factors which bear upon the problem. The factors having the greatest influence are now well recognized as being:

(*a*) The combustion characteristics of the fuel and the nature of the products of partial combustion. The greater the quantity of these highly active substances in the cylinder, the more concentrated the corrosive attack on the exposed surface at the top of the liner, and therefore the greater the rate of wear.

(*b*) The working temperature of the liner surface. Corrosion is greatly assisted by the presence of moisture. If the surface temperature is low enough to permit moisture to linger on the liner during the cycle, chemical action between the offending products and the liner material is encouraged, thereby increasing the rate of wear.

(*c*) The condition of lubrication of the cylinder bore. If the cylinder lubrication is defective, the liner surface is, of course, less well-protected against corrosion, and increased wear results.

The relative effects of these factors will depend largely on the system of combustion and the type of fuel employed. The normal petrol engine is less prone to vagaries in the combustion process than the Diesel engine. Hence as regards cylinder wear the petrol engine is less sensitive to the burning qualities of the fuel than the Diesel engine. Those petrol engines which operate at a high load factor and with a relatively high cylinder temperature show the smallest rate of wear, irrespective of the fuel used. Those which operate under cold conditions and spend a large proportion of their life in running under light load or in starting and warming up show a very high rate of wear, which cannot be reduced by changing the fuel. Typical examples of these two extremes are the aero-engine and the small fishing-boat engine, both of which bear out these observations. In the aero-engine the rate of liner wear is very low. In the small marine engine it is very high.

The Design of Piston and Liner in relation to Cylinder Wear.—Certain features of the piston and ring assembly have been found to give some alleviation in cases of acute cylinder wear. They may be given briefly as follows:

(1) The use of a deep top land. The top land should be made as deep as is mechanically possible, in order to give protection to the top ring and to delay slightly the rise of pressure behind the ring. In this way the squeezing-out action of the ring in removing oil from the liner is postponed until there is more oil to " squeegee " and the ring rides the oil film.

(2) Free circulation of lubricating oil to the ring assembly. In order to wash away both the gum formed by the partial combustion of the lubricating oil and the carbon which is continuously being deposited on the surfaces, it is found vitally important to renew constantly the oil trapped in the ring assembly. This may be done by flooding the lower part of the cylinder bore with oil and by providing the piston with adequate scrapers and relief holes to carry the oil back to the crankcase via the inside of the piston. A brisk circulation of oil is thus maintained and oil stagnation in the ring grooves is avoided.

If gummy substances are allowed to remain in the grooves, the rings become stuck and cease to function as piston rings. " Blow by " the ring face then occurs, destroying the lubrication and increasing the rate of wear.

If carbon deposit is allowed to accumulate in the back of the grooves the rings become packed out. They then become bottle-tight in the cylinder and give rise to a high rate of wear.

(3) The use of a wide top ring on the piston often markedly reduces the rate of wear. The top ring is enabled to carry a larger quantity of oil to the upper limit of ring travel, so that the oil is less easily squeezed out and dispersed when gas pressure comes on the back of the ring.

(4) It is well known that some coarse-grained types of cast iron have the property of storing or holding a certain quantity of oil within the surface. By the use of such material for the piston rings the lubrication of the rings, and of the top ring in particular, may be vastly improved, with a corresponding improvement in cylinder wear. If a very hard liner material of very smooth finish, such as is provided by a chromium-plated surface, is used, it is doubly important that a ring material possessing this oil-holding quality should be provided, otherwise the lubrication of the critical zone may break down altogether, leading to excessive wear and loss of the benefits of a hard non-corroding surface.

CHAPTER X

ENGINES FOR ROAD VEHICLES

Before we deal with specific examples of motor-vehicle engines it will be well briefly to review the duties which these engines have to perform and note along what lines further development is likely to extend.

Some thirty-five years ago the designers and manufacturers of motor-vehicle engines had to concentrate the whole of their attention upon the one crucial problem of producing engines which would run for a reasonable period, under wholly novel conditions, without serious breakdown. This great problem overshadowed all others, and the rapidity with which it was tackled and overcome is one of the triumphs of modern mechanical engineering. Within a space of less than ten years the motor-vehicle engine developed from a fickle and wayward, but very fascinating, toy into a thoroughly reliable machine. Once its reliability had been established and its charm largely exchanged for utility, the subsequent advances were mainly in the direction of refinement and increased power.

As the industry developed, manufacturers were able to provide the same article at greatly reduced cost, and the motor-car has become not only the luxury of the well-to-do but a widely used means of transport displacing the horse vehicle.

During the last ten years designers of motor-car bodies have given much attention to making appeal to the æsthetic taste of the public, and the motor-car has emerged as a fourfold entity, a thoroughly efficient and reliable piece of engineering, an indispensable means of transport in the life of the modern community, a creature of fashion and a victim of taxation.

The first important development in the design of the engine was to provide a more uniform turning moment and to reduce vibration, and therefore noise. The number of cylinders was increased from the original one to four, and then to six and even eight and twelve cylinders. The next developments were in the direction of securing greater silence in operation; these took the form of improvements in the valve gear and the elimination as far as possible not only of vibration of the engine as a whole, but also of vibration of the individual members of the engine. Attention to refinement and the general comfort of the driver and passengers has not stopped here, and elaborate precautions are now taken to isolate engine vibration altogether from the car body by special engine mountings and resilient insulating materials.

At the same time the available speed range has steadily increased. Owing to the inherent irregularity of torque in any type of four-cycle engine the speed range cannot usefully be reduced below a certain minimum, so that progress has taken the form of extending the upper end of the speed range. Developments in this direction have been greatly stimulated by the method of basing the taxation of private vehicles upon the diameter of cylinder used. This basis of taxation has served well, but taxation based on the total cylinder capacity might now be more useful.

With the extended use of motor vehicles for utility purposes the need for development in the direction of fuel economy became insistent, and although much progress has been made, there is still considerable scope for improvement. One of the most important factors controlling fuel consumption is distribution, and it is by the design of induction manifolds giving better distribution, and by the provision of a homogeneous mixture to the cylinders by good carburation and improved methods of charge heating, that improvements in fuel economy have been effected.

The motor vehicle of to-day is called upon:—

(1) To be silent under all conditions of operation.

(2) To be self-contained and as automatic as possible.

(3) To have as wide a range of speed as possible.

(4) To accelerate rapidly from any speed; in other words, it must instantly develop its maximum torque when called upon to do so, irrespective of engine speed.

(5) To maintain a high torque at low speeds, and to do so without detonation or "pinking".

(6) To be reasonably economical in fuel at all loads, more particularly at its average load factor of from 20 per cent to 30 per cent maximum torque.

The performance of any motor-vehicle engine must be considered in reference to the vehicle to which it is fitted. We shall therefore examine briefly one specific instance, namely, a motor-car fitted with a saloon four-seater body, weighing fully loaded, including passengers and equipment, 31¼ cwt. We shall assume that the transmission gear is of an efficient type, that the unsprung weight is low, that the weight is well distributed, and that the chassis generally is as well-designed throughout as the present state of the art will permit.

Unfortunately very few accurate data are available on the exact power required to propel a motor vehicle at different speeds over average roads. Professor Riedler in Germany, and Chase and James in America, have made and analysed a number of dynamometer

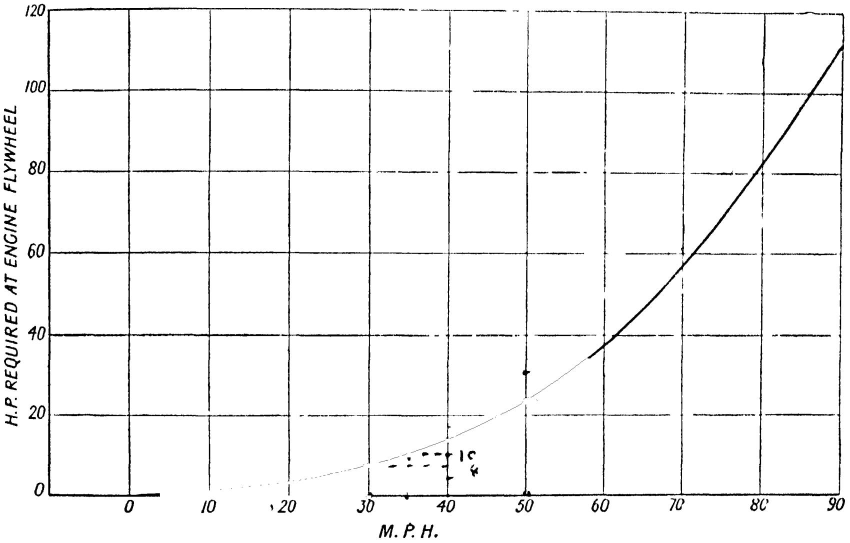


Fig. 150.—Power required at Engine Flywheel to propel Car of 3500 lb. gross weight on the level at varying speeds over average road surfaces

tests with the rear wheels of the car resting on rollers, but these do not always reproduce the conditions exactly. For information on this point we are compelled to fall back to a large extent on tests carried out with accelerometers and on accumulated experience based upon the known performance of the same engine, both on the test-bed and on the road—the latter method, though very unscientific and largely empirical, is probably the most accurate at the present time. The curve in fig. 150 is based on the assumption that rolling resistance is equivalent to a force of 30 lb. per ton weight and that all the losses, including transmission losses, wheel slip, and hysteresis loss with tyres, may be resolved from the empirical formula

$$F = 0{\cdot}002\,AV^2,$$

where F is the equivalent force in lb., A the frontal area in square ft. (assumed 26 sq. ft.), and V the speed in miles per hour. This gives a very close approximation to the brake horse-power required at the engine flywheel to propel such a car at speeds up to 90 M.P.H. Though purely empirical, it is probably reasonably correct.

For such a car two power units will be considered, the first an engine of 2-litre capacity of normal type giving acceleration and hill-climbing capacity such as would be expected from the average medium-sized saloon car of to-day; the second an engine of 3-litre capacity corresponding, in a car weighing 3500 lb., to a high-powered car of the more expensive brand, with a correspondingly more lively road performance. We shall consider both cases and assume that the engines are of normal design with an efficient form of combustion chamber. Further, we shall assume that the engines are designed with a view to low cost of production and ease of upkeep; that they have a reasonably low compression ratio, namely 6 : 1, to render them capable of using inferior fuel without detonation, and generally that they are of orthodox type, but as efficient as possible without resort to features involving either increased cost of production or labour in upkeep.

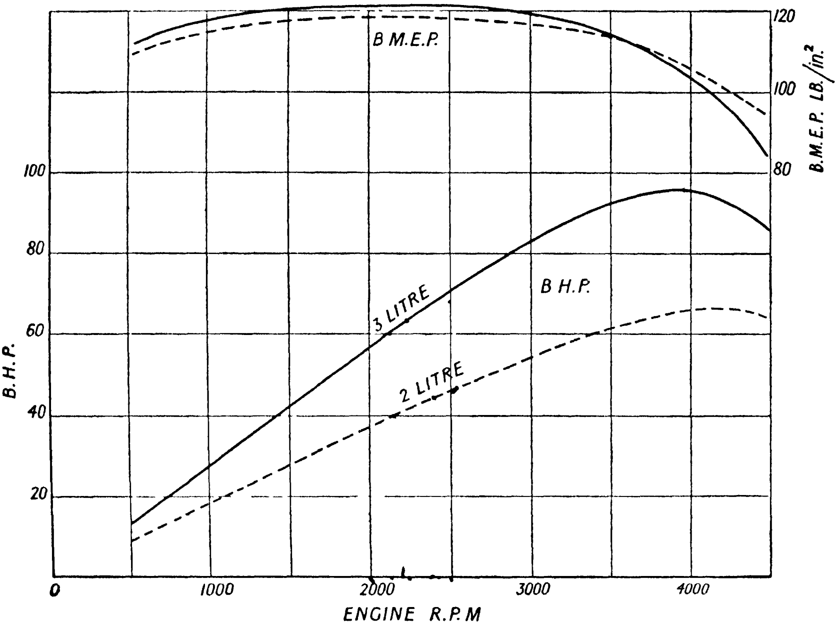


Fig. 151.—Brake Horse-power and Brake Mean Pressure of Normal Two- and Three-litre Engines

Figs. 151, 152 and 153 give the brake horse power and general performance curves which should be obtainable from such engines of two- and three-litre total cylinder capacity.

Next we shall assume that in both cases the top gear ratio is such that the maximum speed on the level is attained when the engine is running at a speed slightly in ~~excess of that at~~ which maximum power is developed; this is desirable both on the grounds of acceleration and in order to reduce the hysteresis losses due to irregularity in the turning moment, but it is not desirable for low fuel consumption—hence the use of over-drive mechanism.

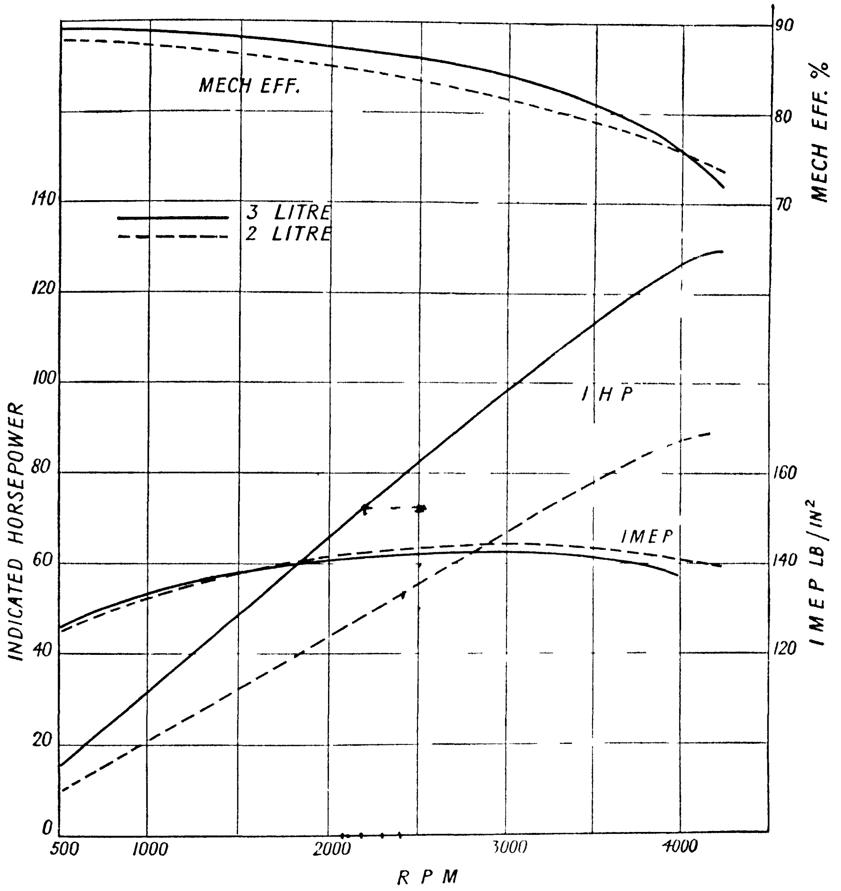


Fig. 152.—Indicated Horse-power, Indicated Mean Pressure and Mechanical Efficiency of Two- and Three litre Engines

We shall assume that four speeds are used and that the ratios in the gear-box for the smaller engine are such that for the third gear the equivalent wheel speed is 67·5 per cent, for the second gear

44 per cent, and for the first gear 26·5 per cent of the speed with top or direct drive. These percentages correspond with average values for the modern gear-box and represent over-all gear ratios of 5, 7·4, 11·3 and 18·9 respectively. A ratio of about 5 : 1 is found to be the most suitable gear for direct drive for the smaller engine, giving a car speed of 20 miles per hour at 1200 R.P.M. with average effective tyre diameters, while for the larger engine a ratio of 4·6 is used, giving, for the same car speed, an engine speed of 1100 R.P.M. The over-all gear ratios in this case then become 4·6, 6·8, 10·5, 17·3. If

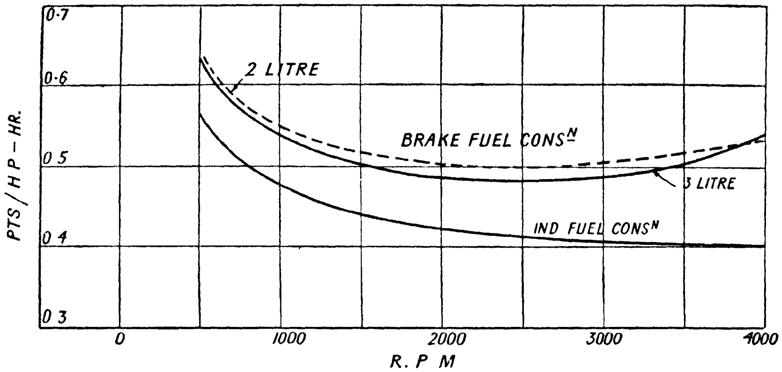


Fig. 153.—Indicated and Brake Fuel Consumption at Full Torque

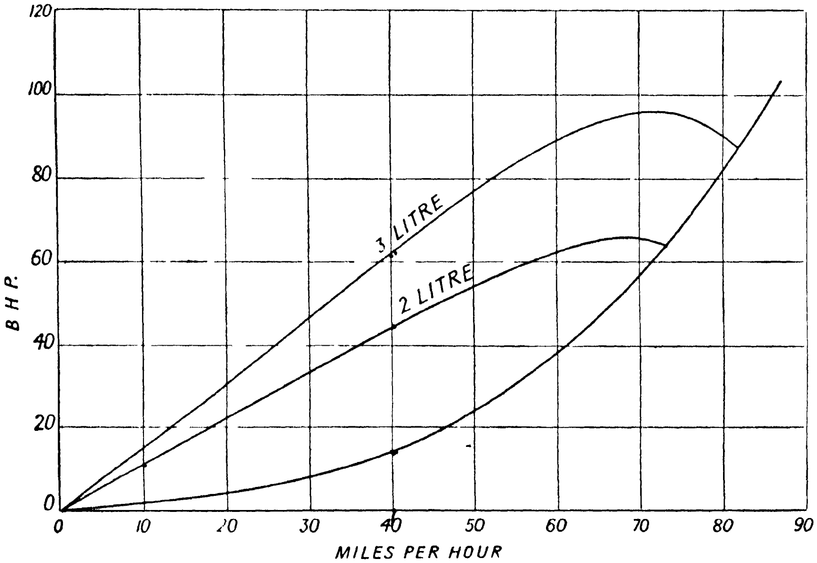


Fig. 154.—Excess-power Curves for Two- and Three-litre Engines

now we plot the power curves of the two engines against the curve of fig. 150 in the above ratios, as shown in fig. 154, we find that the two-litre engine will give a maximum speed on the level of 73 miles per hour and the three-litre of 82 miles per hour. The margin of power at any speed over and above that required to propel the car on the level may be termed the " excess power " available for hill-climbing or acceleration. Figs. 155 and 156 show the excess power

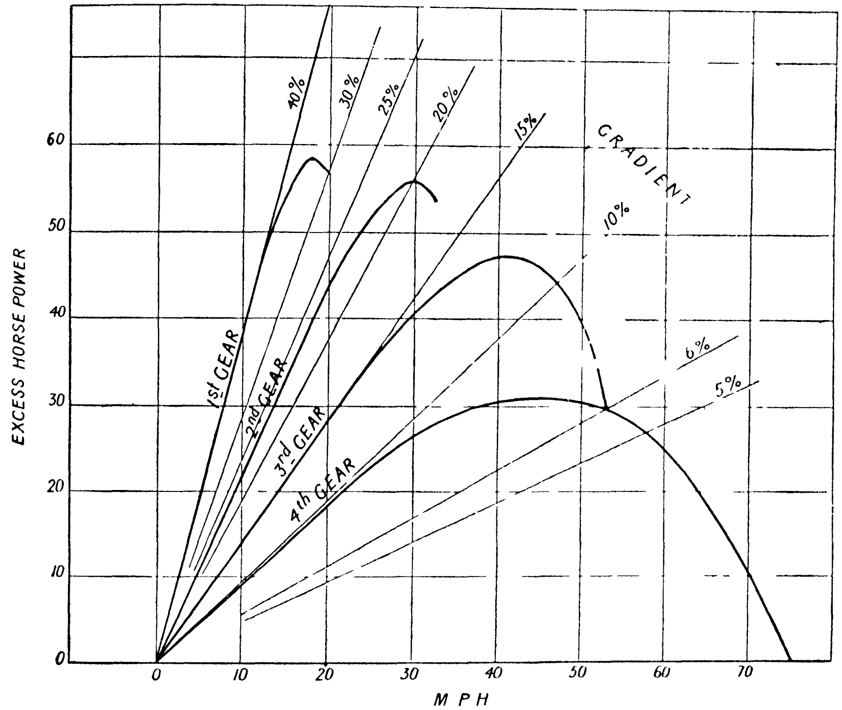


Fig. 155.—Hill-climbing Capacity, Two-litre Engine

curves for the two engines on the four gears, assuming an efficiency as compared with top gear of 94, 96 and 98 per cent for the first, second and third speeds. Strictly speaking, the relative gear losses will be less in the case of the larger engine, but the difference is small and hardly worth taking into consideration. Figs. 155 and 156 show also the gradient which the car will climb on each gear and the speed at which it will climb it without gain or loss of speed.

From the curves shown in these two figures it will be seen that the maximum gradient which the three-litre engine will climb without gain or loss in speed on its fourth or top gear is one of 13·5 per cent, on its third gear the maximum gradient is about 22 per cent, on its second gear 35 per cent, and on its first or bottom gear about 55 per cent. It will be noted that on a gradient of 10 per cent its maximum speed will be the same on either top or third gear, namely, 56 miles per hour.

In the case of the two-litre engine the maximum gradient which the car will climb at a uniform speed on top gear is one of nearly 10 per cent, on third 15 per cent, on second 23 per cent, and on bottom gear about 40 per cent. For maximum speed in this case the gear should be changed from top to third when the gradient exceeds 6 per cent or when the speed has dropped to 53 miles per hour.

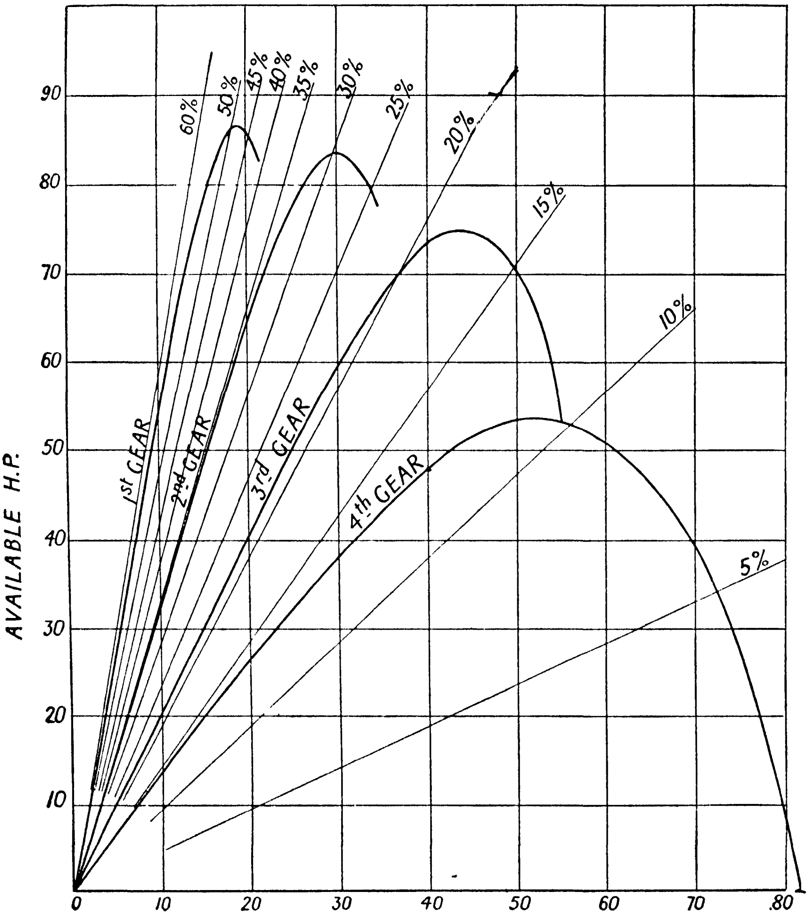


Fig. 156.—Hill-climbing Capacity, Three-litre Engine

Fig. 157 gives the rate of acceleration of the car with two-litre engine, from any speed and on the three gears, assuming that the carburation and distribution are such that the engine will respond instantly and exert its maximum torque immediately the throttle is opened—a condition which is seldom reached in practice. The foregoing curves show the general performance of the car as regards ultimate speed, acceleration and hill-climbing capacity.

We have next to consider the question of fuel consumption and the factors which control it. For this purpose we shall assume that the car will be running always on its top gear and we shall examine the speed range between 10 and 50 miles per hour, which covers the average range of speed over which such a car will operate. For simplicity we shall assume also that the car is running on a level road, although, so far as fuel consumption is concerned, it makes comparatively little difference whether the road is level or undulating provided the gradients are well within the limits which the car can negotiate on top gear and the average speed is not too low. Although when over-running, one does not recover what is lost when climbing, yet this is very nearly compensated by the more favourable load factor when pulling uphill.

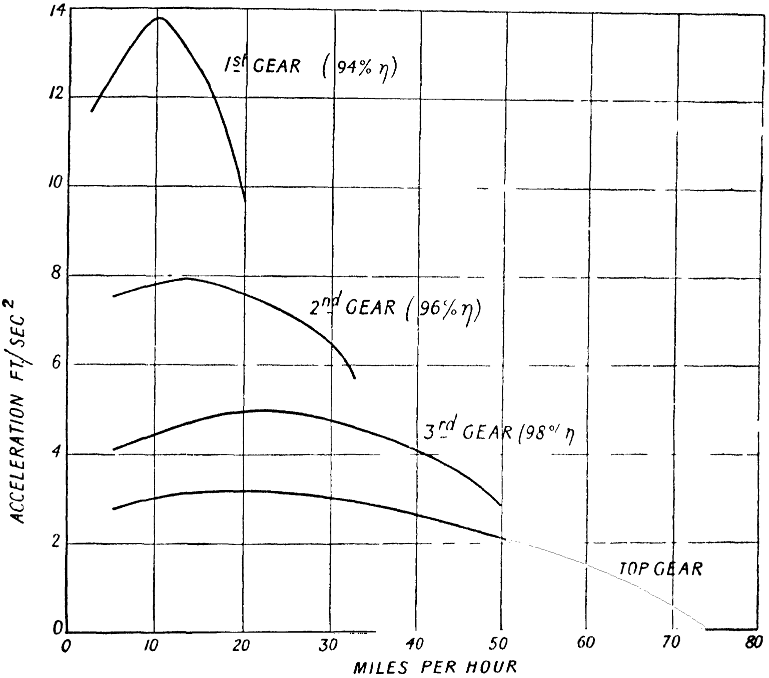


Fig. 157.—Acceleration on various Gears, Two-litre Engine

Fig. 158 shows the load factors for the three- and two-litre engines at speeds ranging from 10 to 50 miles per hour, and the fuel consumption in terms of pints per B.H.P. hour at the corresponding load factors and speeds. These figures are deduced from the mean of a large number of test results upon several engines of the size and class under consideration with carefully adjusted carburettor and ignition settings and a reasonably good distribution system. Fig. 159 shows the fuel consumption in terms of miles per gallon for the two engines at speeds ranging from 10 to 50 miles per hour.

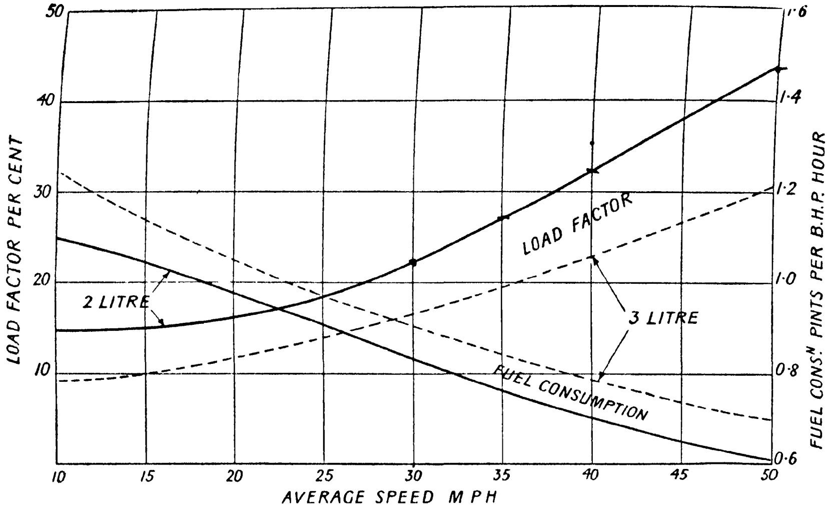


Fig. 158.—Load Factor and Fuel Consumption on Level, Two- and Three-litre Engines

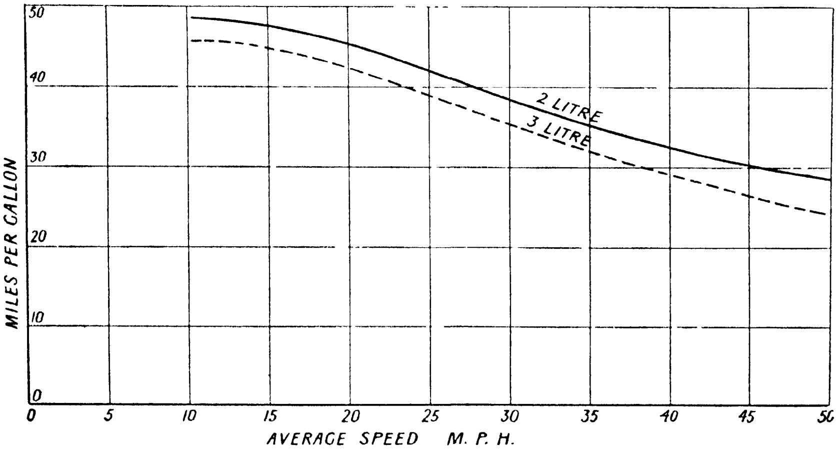


Fig. 159.—Fuel Consumption at various Average Speeds

With normal carburation the consumption per mile is about 10 per cent greater with the larger engine at an average speed of 30 M.P.H. In either case the larger engine will, however, make a better showing if the road is hilly or winding, for the car will then be able to negotiate gradients and corners on top gear which, with the smaller engine, might necessitate a change of gear.

There is another factor which exerts a still more powerful influence upon fuel economy than carburation and distribution, and indeed upon the whole performance of the car, namely, the mechanical efficiency of the engine. This becomes the more important because of the very low load factor at which the engine operates. In the example shown a fairly high mechanical efficiency has been assumed, such as would be obtainable with light reciprocating parts and careful mechanical design.

It is perhaps worth while to consider the case when the car is travelling on the level at a mean speed of, say, 40 miles per hour and observe the influence of the mechanical efficiency of the engine upon fuel consumption. This speed calls for an expenditure of 14 B.H.P. at the flywheel of the engine and corresponds to an engine speed of 2200 R.P.M. in the three-litre and 2400 R.P.M. in the two-litre engine; at these speeds the mechanical efficiencies of the two engines have been taken as 86 per cent and 84 per cent respectively on full torque, but since the internal friction of the engine is independent of the torque, at 14 B.H.P. the mechanical efficiencies will have fallen to 58 per cent and 62 per cent respectively. If now the mechanical losses are doubled owing to poor mechanical design, all other conditions remaining the same, the mechanical efficiencies when the car is driven on the level at an average of 14 B.H.P. will become only 41 and 45 per cent respectively, and the fuel consumption per B.H.P. hour will increase from 0·785 and 0·700 to 0·88 and 0·78 pints per H.P. hour, making allowance for the fact that, owing to the poorer performance as a whole, the load factor will be appreciably higher. We then find that the consumption at a mean speed of 40 M.P.H. in terms of miles per gallon will become 26 and 29·3 as against 29 and 32·5 for the three- and two-litre engines respectively.

From these figures it will be seen that the gain in fuel economy to be obtained by a limited and perfectly possible improvement in mechanical efficiency is a very substantial one. Further, a gain in mechanical efficiency will influence not only the fuel economy, but also the speed and hill-climbing capacity of the car throughout its whole working range. From such considerations we are justified in assuming that of the available means for improvement the most important is that of reducing the internal friction losses of the engine as far as possible, and next in importance are improvements in carburation and distribution.

Unlike engines for other purposes, we may regard the touring-car engine as one which will never be called upon to develop high power, except for very short periods and on special roads such as the Italian *Auto-strada* and German *Autobahnen*, and we have shown that in the engines considered the average load factor under normal running conditions is about 20 to 30 per cent. Expressed in other terms, the average power required at the engine flywheel to propel a touring motor-car under normal conditions at 30–35 miles per hour average speed is approximately 7 H.P. per ton (unladen), while even with the most reckless driving it is almost impossible on an English main road to average 15 H.P. per ton, altogether irrespective of the maximum power of the engine.

Thus the reason for the low load factor of operation of the power unit of a motor vehicle lies in the nature of the work required of it. Owing to the progress made in the provision of more efficient combustion chambers and in general engine design, permitting higher speeds and a greater specific power output, the attainable horse-power for a given engine size and so for a given car weight has increased enormously. It may be said that for the motor-car used as an example, weighing 31¼ cwt., the maximum power output of a two-litre engine 20 years ago was about 35 B.H.P. and the maximum speed of the car on top gear about 55 M.P.H. To-day an engine of similar capacity fitted in a car of similar weight would develop something like 65 B.H.P. and enable a maximum car speed of 75 M.P.H. to be attained.

Although this increased performance is available it cannot be made use of except in very short spurts on favourable stretches of road. A high performance is, however, necessary to meet the demands of the modern user who has been provided with and led to expect a vehicle which will give an ever wider range of performance, better acceleration, higher speeds, and a higher torque at all speeds to provide plenty of excess horse-power for hill-climbing. The greater part of the life of a vehicle is, however, still made up of steady running at a comfortable speed. During the last 20 years this speed has risen by a small amount, from 25–30 to 35–40 M.P.H. The rise has, however, been limited by road and traffic conditions, with the result that only for a very small part of the running life is the extra power now available utilized. Hence, in spite of the higher speed, the average load factor has fallen.

Taking an arbitrary speed, say 35 M.P.H., the load factor 20 years ago was of the order of 45 per cent, whereas to-day at the same speed and for the same car weight and engine capacity the value is about 25 per cent.

The importance of efficient operation over the load range, especially at small throttle openings, cannot, therefore, be over-emphasized.

In such circumstances features having the most devastating effect on fuel consumption are:—

(1) A low mechanical efficiency, in the one application above all others where it should be as high as possible.

(2) A type of combustion chamber giving slow uncertain ignition and combustion at light loads.

(3) Defective carburation and distribution, particularly the latter.

These factors have received too little consideration Until recently development was confined almost solely to the addition of various refinements, to the elimination of noise and general smoothness of operation; such lines of development were, of course, very proper, but there was a tendency to overlook the economic fact that the efficiency of a vehicle as a whole lies in the number of ton-miles it will run per gallon. In too many cases, fuel efficiency appears to have been forgotten entirely in the search for silence in the better-class cars and for low cost of production in the cheaper varieties. The authors use the word " forgotten " advisedly as against " foregone ", for, as has been shown in previous chapters, fuel economy is largely a question of design, and can usually be attained without adding to the cost and without the loss of the desirable features.

The history of engine development has been much the same in all classes of mechanical engineering: first a struggle to attain mechanical reliability, during which stage the engine is a fascinating toy; this is generally followed by a period of intense rivalry in detail refinement to the neglect of other considerations; finally the inexorable laws of economy insist that attention be concentrated on what is really the final test, namely, the amount of work an engine will do on a given quantity of fuel and on a given weight and cost of material. In the case of the motor-car we are in the throes of the third stage of development and are realizing the absurdity of having developed a potentially efficient machine, quiet and well-behaved in operation, only to prevent it giving of its best by poor carburation, casual distribution, and irregularity in spark phasing and spark intensity between cylinders.

As in the case of all new developments which fall into the hands of the lay public, fashion plays a predominant part, and hitherto fashion has called for greater refinement in driving and riding comfort to the neglect of all other considerations.

Just prior to the outbreak of the present war the call for economy in operation was becoming insistent from many quarters. The war has placed a new emphasis on economy and will focus attention more and more on those features in design which affect fuel consumption. Any lack of uniformity, whether in mixture strength, mixture quantity, or spark phasing, raises the general level of mixture strength needed by the engine to ensure regular firing on all cylinders, thereby increasing the fuel consumption. In most cases much remains to be done to bring about the uniformity of operation between cylinders which is vital to good fuel economy. When good distribution has been achieved, it remains to provide a fuel mixture which maintains a steady fuel/air ratio over the speed range and over the whole range of throttle opening: no easy problem.

The Vauxhall "Fourteen" Engine.—The Vauxhall "Fourteen" engine illustrated in figs. 160 and 161 is an excellent example of modern practice in automobile engine design. The author is greatly indebted to Mr. Alex Taub for leave to publish the following particulars of this engine.

It may be regarded as a typical example of an up-to-date car engine of the best type, designed to meet as far as possible both the dictates of fashion and the requirements of to-day. It has six cylinders, each of 61·5-mm. bore and 100-mm. stroke, giving a total cylinder capacity of approximately 1·78 litres, and is designed to drive a *de luxe* saloon car. In the interests of high road performance the use of an unduly capacious body has been avoided, thus curtailing weight and wind resistance. A further reduction in vehicle weight is rendered possible by the use of an integral all-steel construction, with the result that the curb weight of the complete car, fully charged with petrol, lubricating oil, and cooling water, is only 2500 lb.

The engine develops just over 48 B.H.P. at a speed of 4000 R.P.M. The maximum brake torque expressed in terms of brake mean pressure on the pistons is 112 lb. per square inch, developed at a speed of 1750 R.P.M.

Particular care has been taken to reduce as far as possible the internal friction losses. The combustion chamber has been designed to give smooth running coupled with high performance.

The general appearance and the position of the accessories are shown in figs. 160, 161, which give photographs of both sides of the engine. The cylinders are cast in one block in chromidium cast-iron integral with the upper half-crankcase, the water jackets extending the full length of the cylinders. Stiffening webs inside the crankcase give a very rigid construction and support the main bearings.

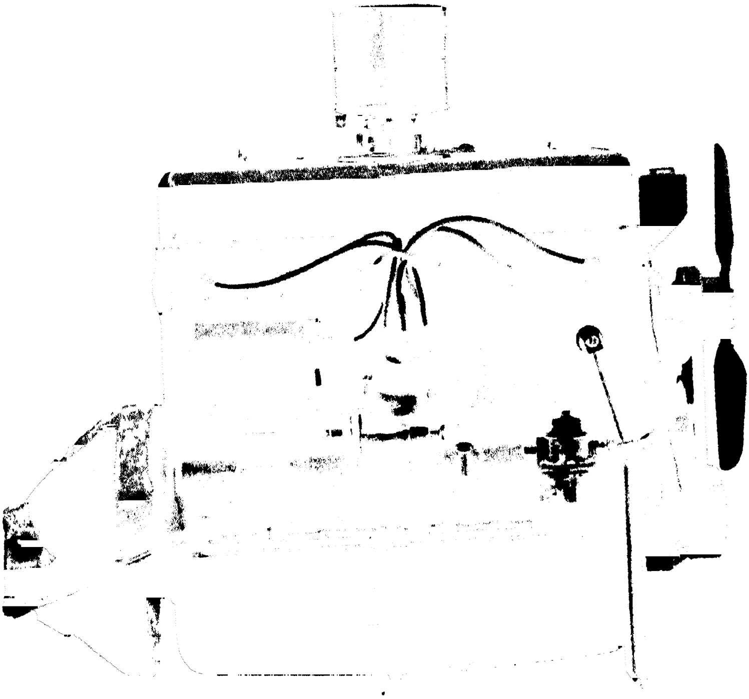


Fig. 160.—Vauxhall "Fourteen" Engine

The cylinder block has a common detachable head carrying overhead valves operated by push-rods.

The crankshaft is supported in four white-metal-lined steel shell bearings and is drilled for forced lubrication to all main and crank-pin bearings.

The pistons are in aluminium alloy and are of trunk form; the skirts are oval-ground.

The gudgeon-pins are clamped in the small end of the connecting-rod, which is split for this purpose.

The total reciprocating weight in each line is 1·25 lb., while the total weight of the rotating part of the connecting-rod and its bearing assembly is 1·11 lb. The inlet valves have a port diameter of 1·25 in. with a lift of 0·34 in. The port diameter and lift of the exhaust valve are respectively 1·125 and 0·335 in.

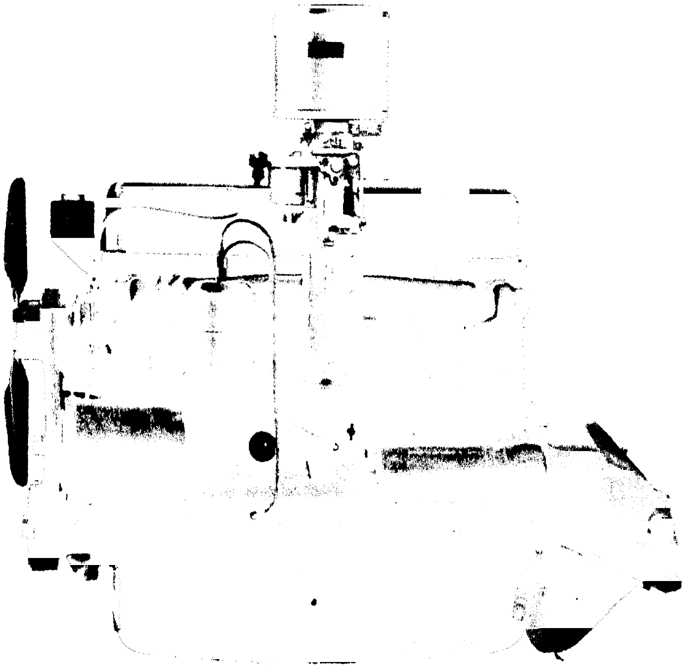


Fig. 161.—Vauxhall "Fourteen" Engine

The crankshaft has four bearings, of diameter and width as follows:—

	Main Journal Bearings
Front bearing	$2\frac{1}{16}''$ dia., width $1\frac{19}{32}''$
Front intermediate	$2\frac{1}{8}''$ dia., width $1\frac{11}{64}''$
Rear intermediate	$2\frac{3}{16}''$ dia., width $1\frac{7}{32}''$
Rear bearing	$2\frac{1}{4}''$ dia., width $1\frac{13}{16}''$
	Big End Bearings
Pin diameter	$1\frac{7}{8}''$ dia., width $1\frac{1}{32}''$ effective

The ratio of the piston area to the projected crank-pin area is therefore 2·4:1.

The general performance of the engine, with a compression ratio of 6·75 : 1 and with wide-open throttle over a speed range from 500 to 4000 R.P.M., is shown in fig. 162, from which it will be seen that a brake mean pressure of 112 lb. per square inch is obtained at a speed of 1750 R.P.M.

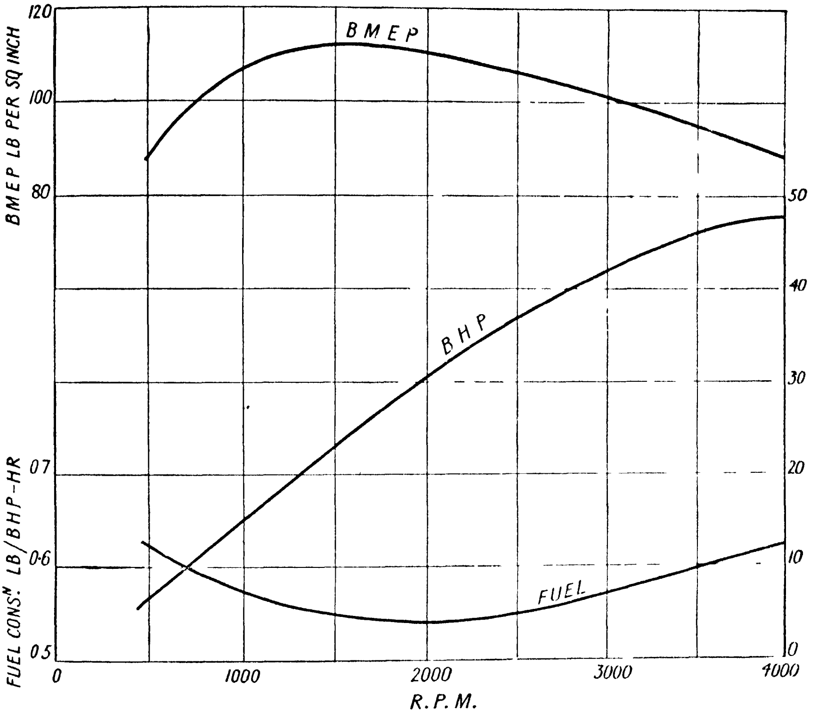


Fig. 162.—Vauxhall "Fourteen" Engine. Brake Horse-power, Brake Mean Pressure, and Fuel Consumption Curves

Fig. 163 (p. 308) gives the mechanical efficiency and the indicated M.E.P. The mechanical efficiency is well maintained throughout the speed range, a feature which enables a good brake torque to be developed up to the limit of speed.

Fig. 164 (p. 308) shows a curve of fuel consumption in miles per gallon plotted against road speed, and also a curve giving the road load in horse-power required to propel the vehicle at each speed on the level. The method of carrying out this test is to take a series of part throttle readings at various speeds with the engine adjusted to give the power output required for the particular speeds at which

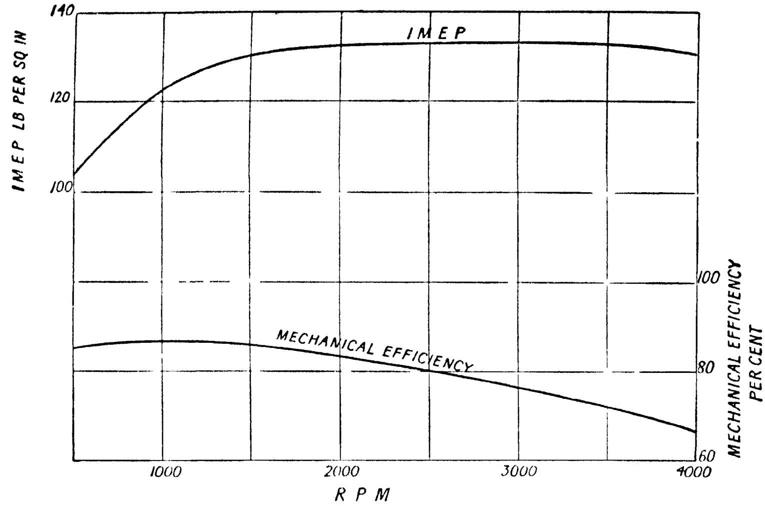


Fig. 163.—Vauxhall " Fourteen " Engine. Indicated Mean Pressure and Mechanical Efficiency

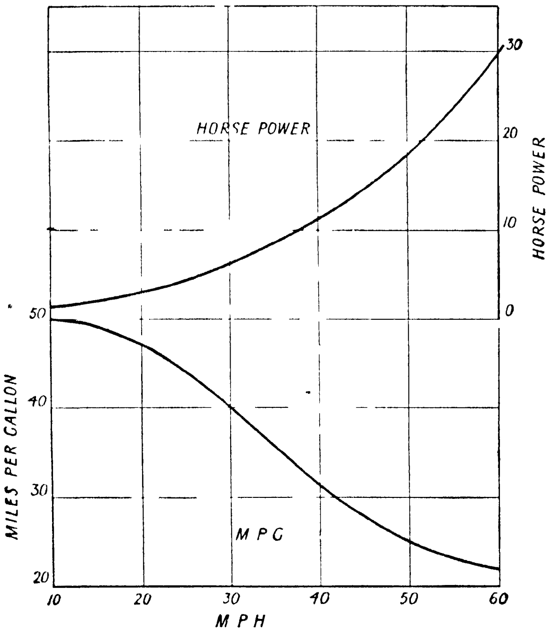


Fig. 164.—Vauxhall " Fourteen " Engine. Road Load Curve and Fuel Consumption

readings are taken. The resulting curve represents the fuel consumption which may be realized on the level, assuming a steady speed to be maintained throughout.

The engine and the three-speed gear-box form one unit. The gear-box has a synchromesh engagement mechanism and silent helical gearing for all three forward ratios. The ratios correspond to road speeds of 4·79, 10 and 16·4 miles per hour at an engine speed of 1000 R.P.M.

The maximum road speed on the level is 70 M.P.H., corresponding to an engine speed (without slip) of approximately 4250 R.P.M. The fuel consumption, maintaining an average speed of 30 M.P.H., is about 30 miles per gallon.

The use of a down-draught carburettor having modern fittings such as an accelerator pump and part throttle economy device is worthy of note. A conspicuous feature (see figs. 160 and 161) is the combined air filter intake silencer and flame arrester, embodying metallic wool soaked in oil.

Sleeve-valve Engines.—For motor-car engines where the need for silence is great, the use of sleeve valves in place of the ordinary poppet valves appears very attractive. Such valves have the following advantages:—

(1) Their action is, or should be, noiseless.

(2) Their employment permits of the use of the best possible form of combustion chamber with the sparking plug centrally situated; hence the indicated efficiency should be high.

(3) They impose no arbitrary speed limit, as they are positively controlled and operated.

(4) They require less attention than poppet valves and cannot readily be thrown out of tune by misuse.

The objections are:

(1) The heat flow to the cooling water is necessarily somewhat restricted, though this is not of much moment in the case of motor-car engines, more especially when a single sleeve is used.

(2) Unless the sleeve is given an abnormally long stroke the effective port area is somewhat restricted.

(3) The sleeve or sleeves, having a large rubbing surface, necessarily entail a higher friction loss, more particularly when a long stroke is used.

(4) It is possible only to operate the sleeve from one side unless the whole of the operating mechanism is duplicated, which involves

excessive mechanical complication and introduces grave difficulties in the way of mechanical synchronization.

More recently the intensive development of the single sleeve has shown that with this form of valve the objections outlined above have to a large extent lost their force, mainly on account of improvements in mechanical design and of the use of suitable materials and running clearances. The development of the single-sleeve engine is given further consideration in a later chapter.

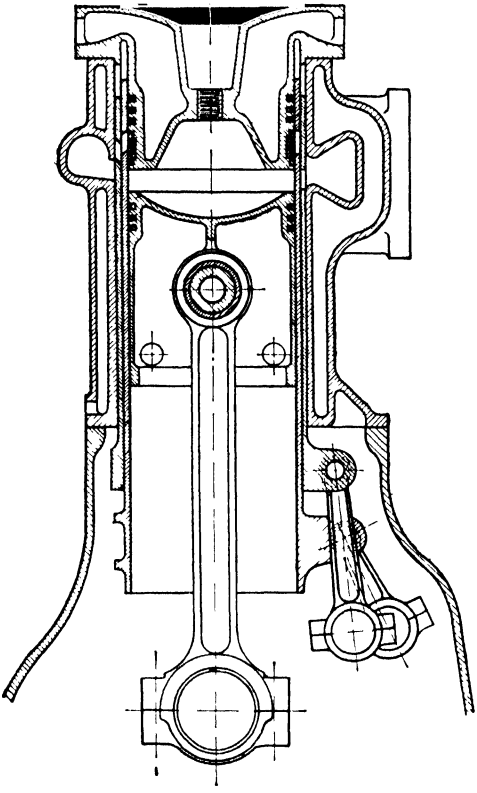


Fig 165

In a four-cycle engine the sequence of operations is such as cannot be fulfilled by a plain reciprocating sleeve; hence it is necessary either to employ two concentric reciprocating sleeves, as in the Daimler Knight engine, or a single sleeve with a combined reciprocating and rotary motion, as in the Burt engine. A plain rotating sleeve is unsatisfactory, since a reciprocating motion of some sort is essential to prevent scoring of the sleeve and cylinder wall. It is preferable also that the whole of the inner surface of the sleeve shall, at every cycle, be scraped either by the piston or the fixed cylinder head, in order to prevent the formation of shoulders due to wear or carbon deposits.

The use of two concentric and reciprocating sleeves has the advantage that their mechanical operation is somewhat simpler, but

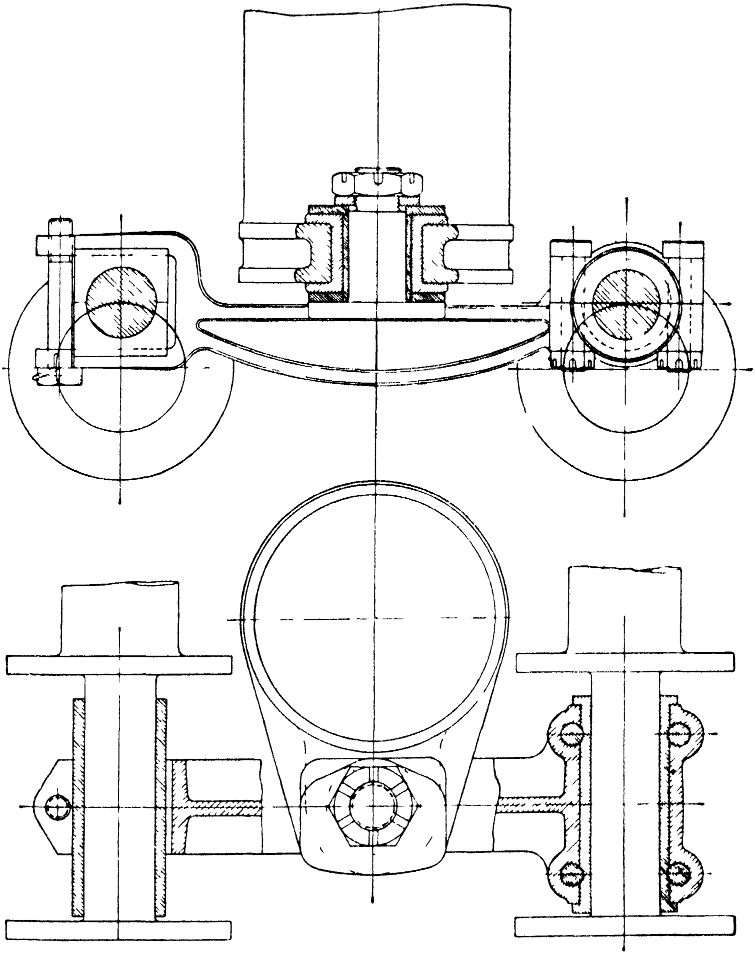
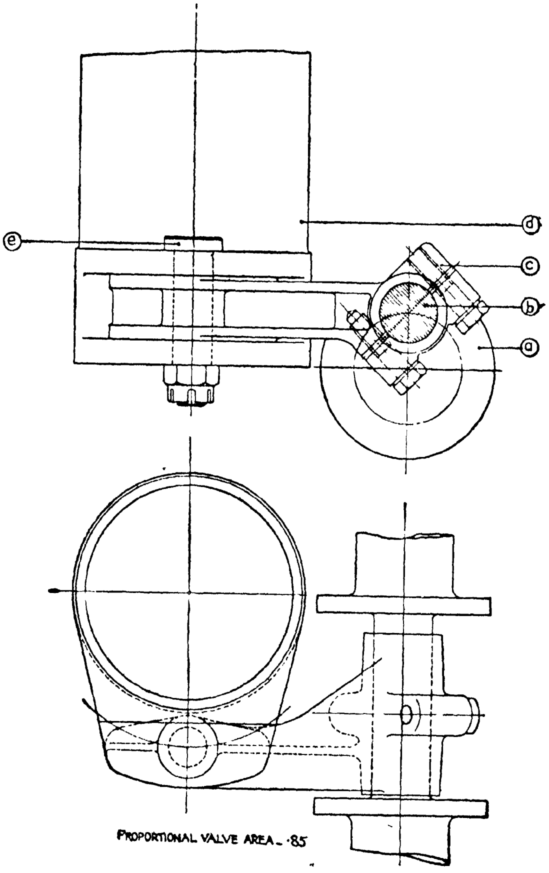


Fig 166

it is very difficult to see what further advantage they can possess. The chief fault of the sleeve-valve engine, namely, the difficulty of disposing of the heat from the piston, is greatly accentuated when two sleeves are used, as also the friction loss, which is no small item and a particularly objectionable one where the load factor is light, as in a motor-car engine.

In fig. 165 is shown diagrammatically the operation of a double-sleeve engine, from which it will be seen that the sleeves are actuated from a half-speed crankshaft connected by short rods to points at the side of each sleeve.

Figs. 166–169 show various alternative methods adopted by Burt for operating a single-sleeve valve. The method shown in fig. 166



is that used in the Picard Pictet cars and is, in the author's opinion, attractive from a mechanical point of view, but it is necessarily somewhat costly. In this arrangement two half-speed crankshafts are employed and the sleeve is operated from the centre of a coupling-rod connecting these shafts. It will be seen that one end of the coupling rod is connected directly to one crankpin and the other to a second crankpin, but through the medium of a sliding block with a small amount of end play. The sliding block allows for any slight errors in synchronism as between the two half-speed shafts.

Fig. 167 shows a similar method of operation in which only one half speed shaft is used. This is said to work well in practice, but is clearly inferior mechanically to that shown in fig. 166.

Another very attractive form is that shown in fig. 168, in which a ball-and-socket joint is used. This form has the advantage of being considerably lighter and more compact; also it is probably the least expensive and the most accessible. Fig. 169 shows an earlier form used in the Argyll cars, in which a reciprocating plunger is used in place of the ball socket. As in the forms 166, 167 and 168, this necessitates the use of a pin in the sleeve, and therefore both increases the radius of operation and the effective weight of the sleeve.

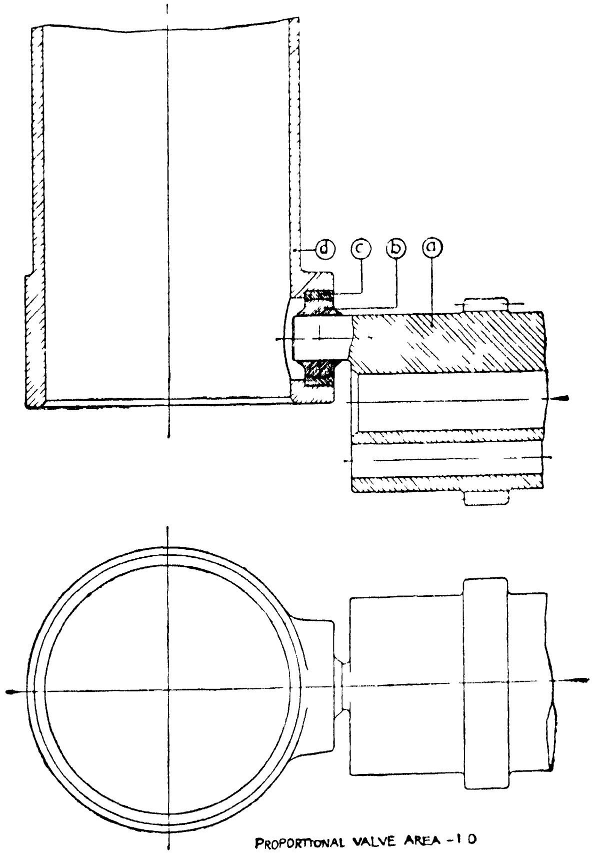


Fig 168

The author is greatly indebted to Mr. Burt for the following information and particulars as to the determination of port areas, &c., when a single sleeve is employed.

Calculation of Ports.—The special shape of port is adopted to give a maximum area of opening with the minimum of sleeve travel.

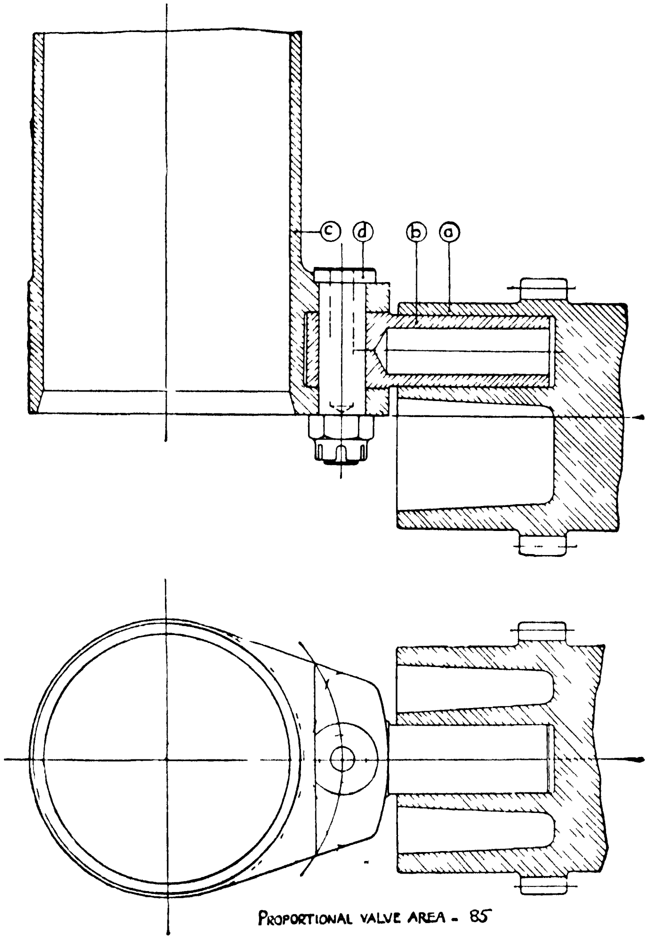


Fig 169

Fig. 170 shows typical ports, *a* being the ideal shape, *b* the same port with corners rounded off to avoid interference when sleeve-valve port is passing down between two cylinder ports. The straight flank port *c* is usually adopted, as it is a better manufacturing proposition, although slightly smaller in area than type *b*, for a given valveshaft stroke.

It is necessary to fix the following particulars before calculating single-sleeve valve ports:—

A = Arrangement and number of ports.
D = Outside diameter of sleeve valve in inches.
C = Distance from axis of sleeve valve to axis of pivot-pin or ball-and-socket coupling in inches.
T = Throw of sleeve crank in inches.
V = Engine timing.

The greater the number of ports the smaller the sleeve-valve crankthrow for a given area of valve opening, thereby reducing the over-all dimensions of valve-driving mechanism and height of engine, but complicating the coring of water-passages in the cylinder castings and increasing the port cutting time. The fewer the number of ports for a similar area the greater the valve crankthrow and over-all dimensions, but the coring is simplified with an attendant reduction of port cutting time.

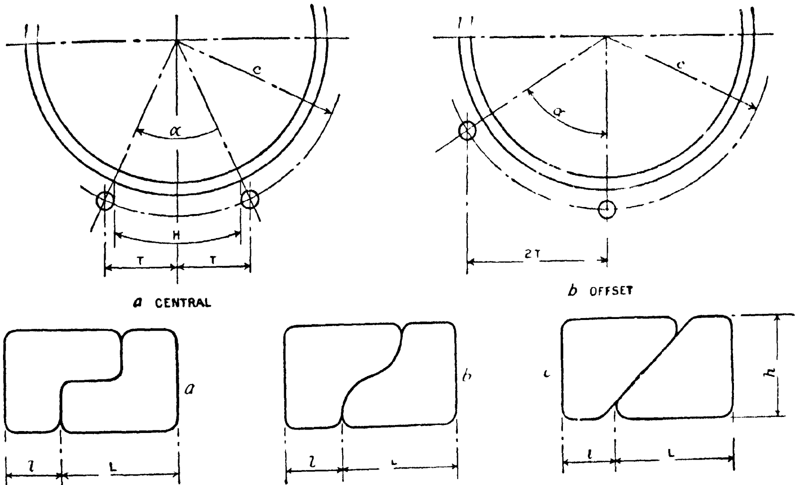


Fig. 170 Angular Travel

The maximum inlet opening areas obtainable with various port settings are given in fig. 171, while fig. 172 illustrates in proportion several settings. It will be noticed that a "double purpose" port—that is, one which acts alternately as inlet and exhaust—is included in each setting. This is advisable where maximum openings are desired; two single ports with a wall between would obviously use up more of the sleeve-valve circumference than a single "double purpose" port.

The sleeve valve is usually centrifugally cast of good quality grey iron, and for engines up to $2\frac{3}{4}$-in. diameter bore is made ·1 in. thick, while for engines of $4\frac{1}{4}$-in. diameter bore ·125 in. thick is quite satisfactory. D can readily be obtained by adding twice the sleeve-valve wall thickness to the cylinder bore diameter, while C, which should be kept as small as possible, is generally ·575D when the ball-and-socket type of drive is used.

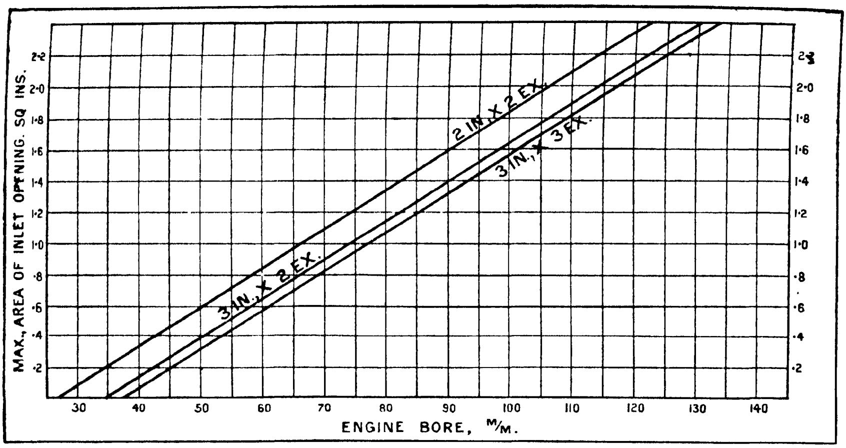


Fig. 171.—Maximum Inlet Valve Opening. Various Port Settings

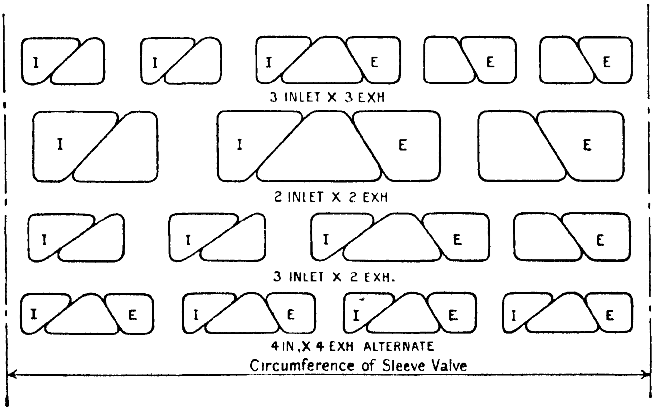


Fig. 172.—Various Port Settings

The throw of sleeve crank T is obtained from the number of ports in the cylinder, as given by the setting adopted, and the dimensions D and C.

$$T = \frac{\pi D \times \cdot 575D}{D(2\text{ No. of ports}) - 1} = \frac{1{\cdot}8D}{(2\text{ No. of ports}) - 1}.$$

This gives the maximum throw possible with this type of actuation and may be reduced within limits according to the type of engine under consideration. Maximum throws for various settings and engine bore diameters are given graphically in fig. 173.

The valve timing V has now to be settled, and in common with poppet valves it varies according to the type of engine.

The actual calculation of ports will be best understood by taking an example. Assume a high-speed engine of 68-mm. bore, the desired maximum opening area of the inlet ports being 1·0 sq. in. approximately, so that we have

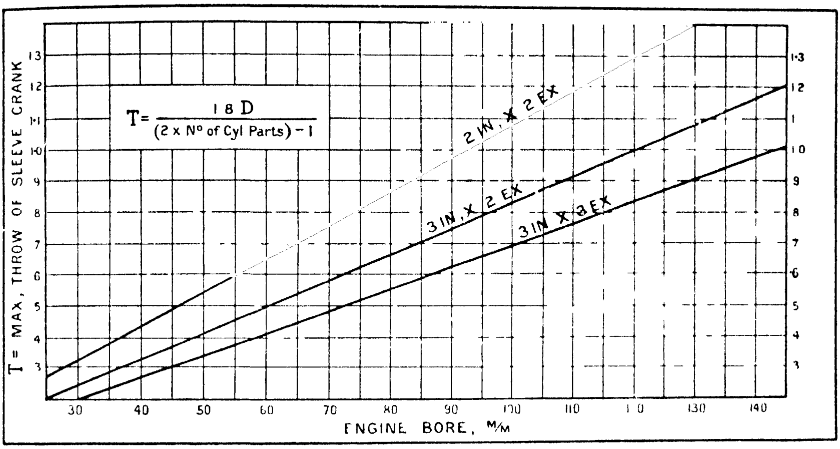


Fig. 173.—Maximum Sleeve-valve Crankthrow. Various Port Settings

A = 2 inlet × 2 exhaust, for it will be seen from fig. 171 that this setting gives the required area.

D = 68 mm. + ·2 in. = say 2·9-in. diameter.

C = ·575 × 2·9 = 1·66 in., say 1·65 in.

$$T = \frac{1{\cdot}8 \times 2{\cdot}9}{(2 \times 4) - 1} = {\cdot}74 \text{ in., say } {\cdot}7 \text{ in.}$$

a = Angular travel of sleeve valve (see fig. 170).

$$\sin \tfrac{1}{2}a = \frac{T}{C} \text{ when T is central} = \frac{{\cdot}7}{1{\cdot}65} = {\cdot}42424, \text{ or } \tfrac{1}{2}a\ 25^\circ\ 6';$$

$$\therefore a = 25^\circ\ 6' \times 2 = 50^\circ\ 12'.$$

When T is offset $\sin a = \dfrac{2T}{C}$.

Referring to fig. 174:

$$H = \text{Horizontal travel} = \frac{a\pi D}{360} = \frac{50^\circ\ 12' \times 3{\cdot}1416 \times 2{\cdot}9}{360} = 1{\cdot}275''.$$

L = Length of port = H − cover = $1{\cdot}275'' - {\cdot}05'' = 1{\cdot}225''$.
Minimum cover = $\cdot04''$. This is usually arranged so as to bring L to an even figure.

W = Minimum space between ports (this does not apply to alternate ports as shown in setting *d*, fig. 172) = H + cover = $1{\cdot}275'' + {\cdot}05'' = 1{\cdot}325''$.

h = Height of ports = $T + (T \sin\beta) = {\cdot}7 + ({\cdot}7 \sin 22\tfrac{1}{2}^\circ) = {\cdot}97''$.

4 ports at 1·225″ long = 4·9″
2 spaces at 1·325 „ = 2·65″
2 spaces at ·775 „ = 1·55″

Circumference of sleeve valve = 9·1″.

$$li = \text{Inlet port tail} = \frac{H}{2} - \left(T \sin\delta \frac{{\cdot}5D}{C}\right)$$

$$= {\cdot}6375 - \left({\cdot}7 \sin 7\tfrac{1}{2}^\circ \times \frac{1{\cdot}45}{1{\cdot}65}\right) = {\cdot}6375 - {\cdot}08 = {\cdot}5575''.$$

$$le = \text{Exh. port tail} = \frac{H}{2} + \left(T \sin\nu \frac{{\cdot}5D}{C}\right)$$

$$= {\cdot}6375 + \left({\cdot}7 \sin 15^\circ \times \frac{1{\cdot}45}{1{\cdot}65}\right) = {\cdot}6375 + {\cdot}1595 = {\cdot}797''.$$

With straight-sided ports the flank angle can be solved as follows (see fig. 174):—

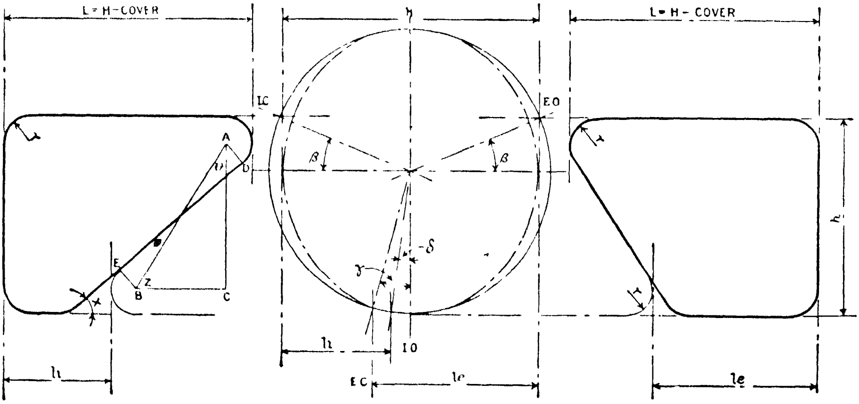


Fig. 174

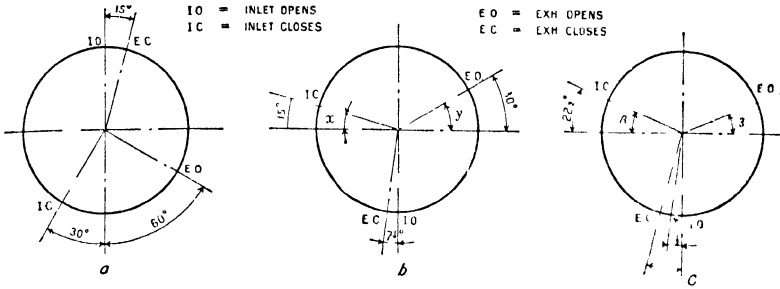
$$AC = h - 2r, \qquad \tan\theta = \frac{BC}{AC}.$$

$$r = \text{corner radius usually } 1/8'',\ AB = \frac{BC}{\sin\theta}.$$

AD and EB $= r$, $\sin \phi = \dfrac{r}{\frac{AB}{2}}$.

BC $= L - (l + 2r)$, $z = 90° - \phi$.

Flank angle $= X = z - \phi$.



CRANKSHAFT TIMING

VALVESHAFT TIMING UNBALANCED

VALVESHAFT TIMING BALANCED 7½"CRANK PIN ADVANCE

Fig. 175.—Timing Diagrams

In fig. 175 the crankshaft timing diagram for the example worked is shown at *a*, while the same timing transferred to the valveshaft is indicated at *b*, the crankshaft being at TDC, while the valveshaft

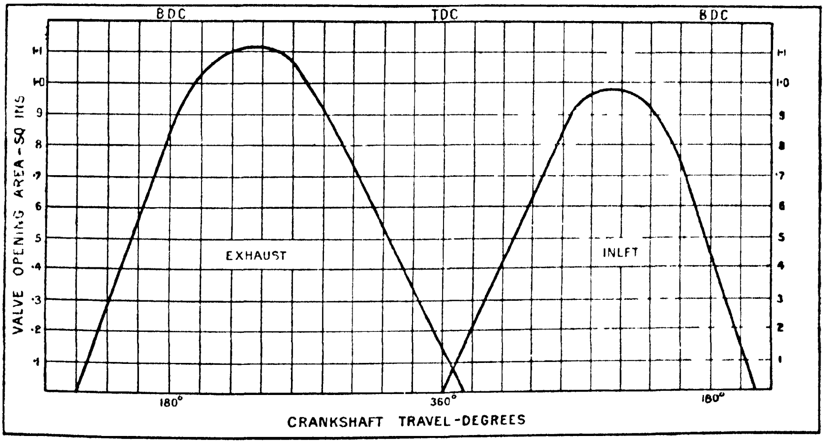


Fig. 176.—Valve-opening Diagram. 68-mm. Bore Engine, C = 1·65″, T = ·7″

crankpin is at BDC. It is obvious that with this valveshaft setting a relatively large opening to exhaust would be obtained owing to the greater height of exhaust port. To overcome this, and in order to make the machining of ports a simpler operation, inlet and exhaust ports are made of equal height. This is made possible by the setting of valveshaft crank-pin in advance of its BDC and in relation to the crankshaft until the angles x and y are equal. The amount of offset is found by $\frac{x+y}{4}$, and the result for the example taken is shown in fig. 175 at c, the corresponding port-opening diagram being as shown in fig. 176.

Fig. 177 shows an example of a six-cylinder motor-car engine of 81·5 mm. bore and 124 mm. stroke in which single-sleeve valves actuated by the ball-and-socket mechanism are employed; while figs. 178 and 179 show a remarkably neat design of motor-cycle engine built by Messrs. Barr & Stroud, and incorporating the same features.

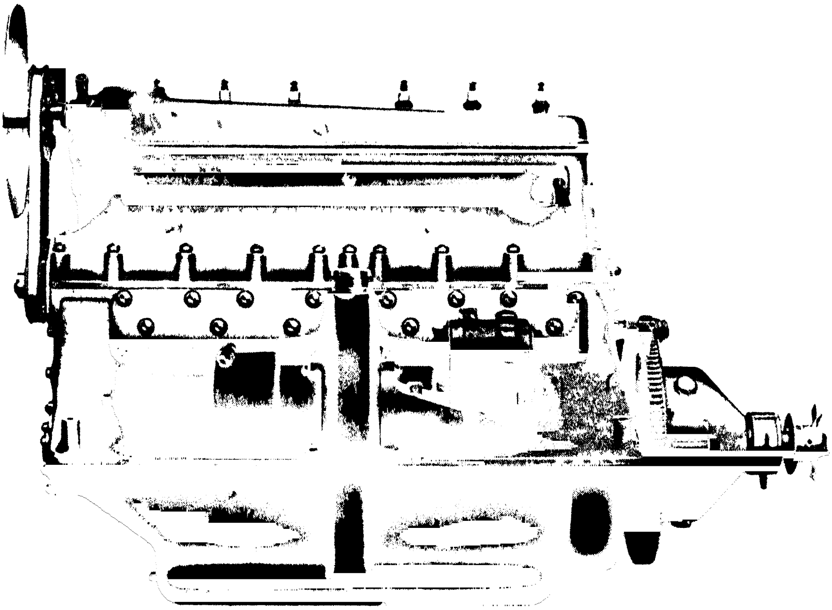


Fig. 177.—Six-cylinder Vauxhall Single-sleeve Valve Engine

Racing Cars.—The practice of motor-car and cycle racing has been one of the most valuable stimulants to the design of efficient internal-combustion engines, for the racing engine operates under conditions of severity such as are met with in no other field, with the result that weaknesses which would develop in the course of years under ordinary conditions of service are shown up in as many minutes. The rapid progress which the high-speed internal-combustion engine has made during recent years is due to the stimulus of motor racing, and that to an extent which few people fully realize.

In the production of a racing engine the designer has full liberty to employ every means known to him to obtain the highest possible power output regardless of any other consideration, except that during more recent years it has become the practice to limit the cylinder capacity of racing engines, a restriction which has proved of undoubted benefit.

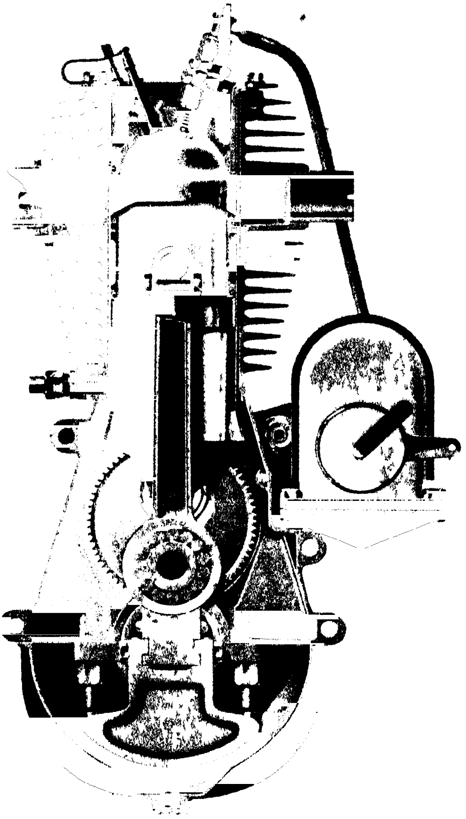


Fig. 178

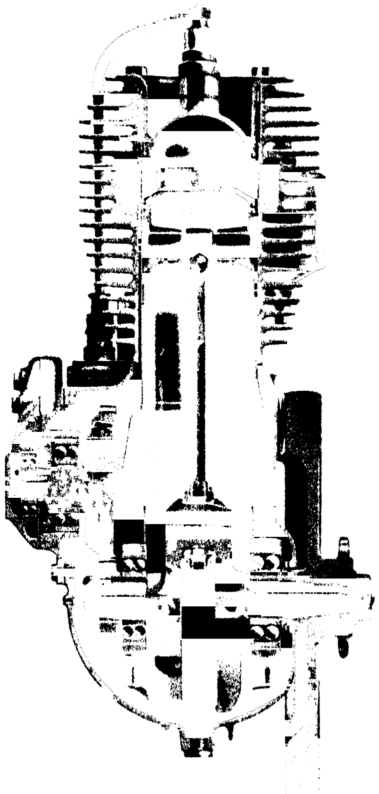


Fig. 179

Barr & Stroud Engine with Burt Single Valve

There is a popular impression that because the racing engine no longer resembles the actual article used in touring cars its value, from an educational point of view, has been lost; this, however, is a sheer fallacy: the racing engine operates on the same cycle and under the same conditions, except that they are much more severe, as the ordinary touring-car engine, and the lessons learnt from its behaviour are just as applicable to the intelligent designer as though the engines were identical.

From an educational point of view it is probably desirable that the racing engine should differ from the touring model, for by its difference—

(1) Higher speeds and therefore more strenuous conditions of test are obtained.

(2) The racing engine of to-day is providing lessons for the future also, and not only for the immediate present.

Again, there is a popular but wholly mistaken belief that the racing motor-car engine, though powerful, is not " efficient ", and that since fuel economy does not enter into consideration such engines teach us nothing about this important question. For an engine to be powerful it must be efficient in every respect—that is to say, it must convert the highest possible percentage of the heat energy available from the combustion of every pound of air into useful work at the flywheel. If, as may sometimes be the case, the air is supersaturated with fuel, that is the carburettor's not the engine's fault, for with good carburation the racing engine will show the highest possible thermal efficiency reckoned on the fuel also.

The engine illustrated in fig. 180 is one of several constructed by Messrs. Vauxhall Motors Ltd. for their racing cars for the 1922 season. It is of three-litres capacity, and develops, the author believes, the highest power output ever yet obtained from an engine of this size. The main features aimed at in the design of this engine were:

(1) To obtain the maximum possible thermal efficiency, with a view to getting the utmost possible power output from the available air.

(2) To ensure the maximum of structural rigidity.

(3) To avoid crankshaft torsional vibration at any speed of which the engine was capable.

(4) To obtain the highest possible mechanical efficiency.

(5) To obtain a high volumetric efficiency.

(6) To provide a form of connecting-rod big-end bearing which should be capable of withstanding continuous running at a mean speed of well over 4000 R.P.M.

The steps taken to meet these conditions were:—

(1) In order to obtain the highest possible thermal efficiency the combustion chamber was made of the shallow pent-roof type with the sparking fitted centrally in the cylinder head. The maximum

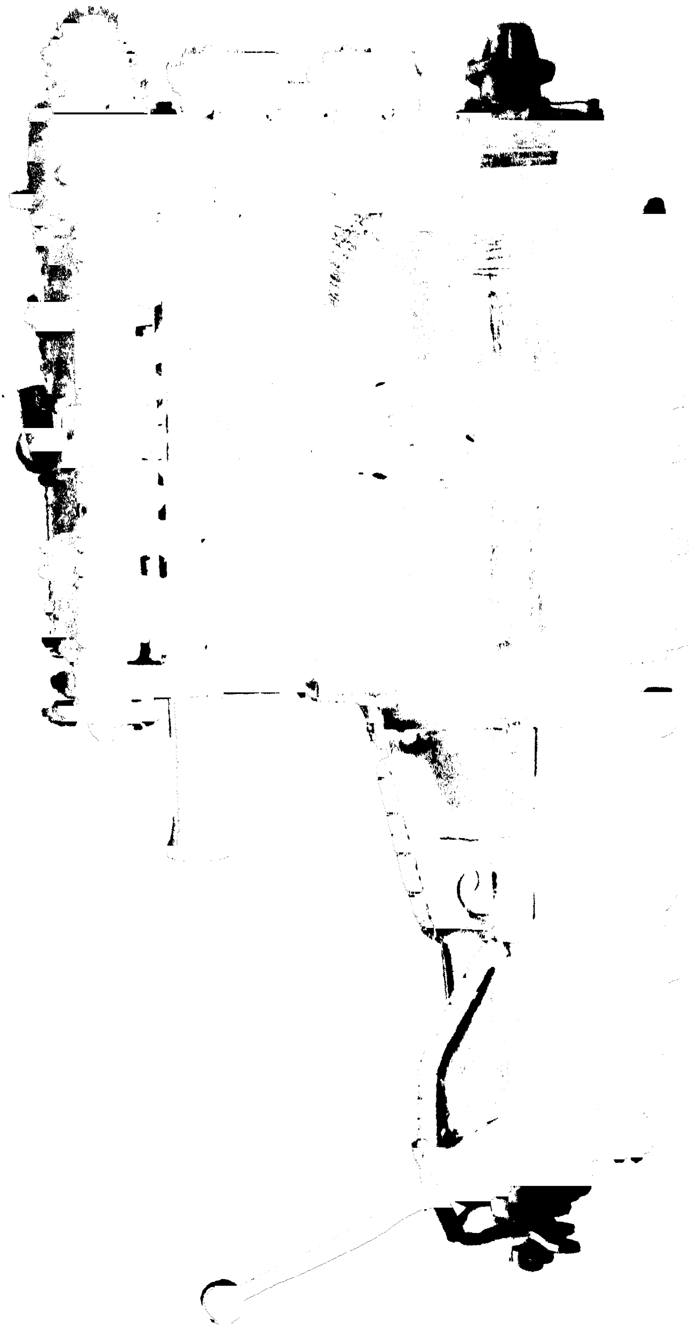


Fig. 180.—Vauxhall Three-litre Racing Engine

Fig. 181 Vauxhall Three litre Racing Engine

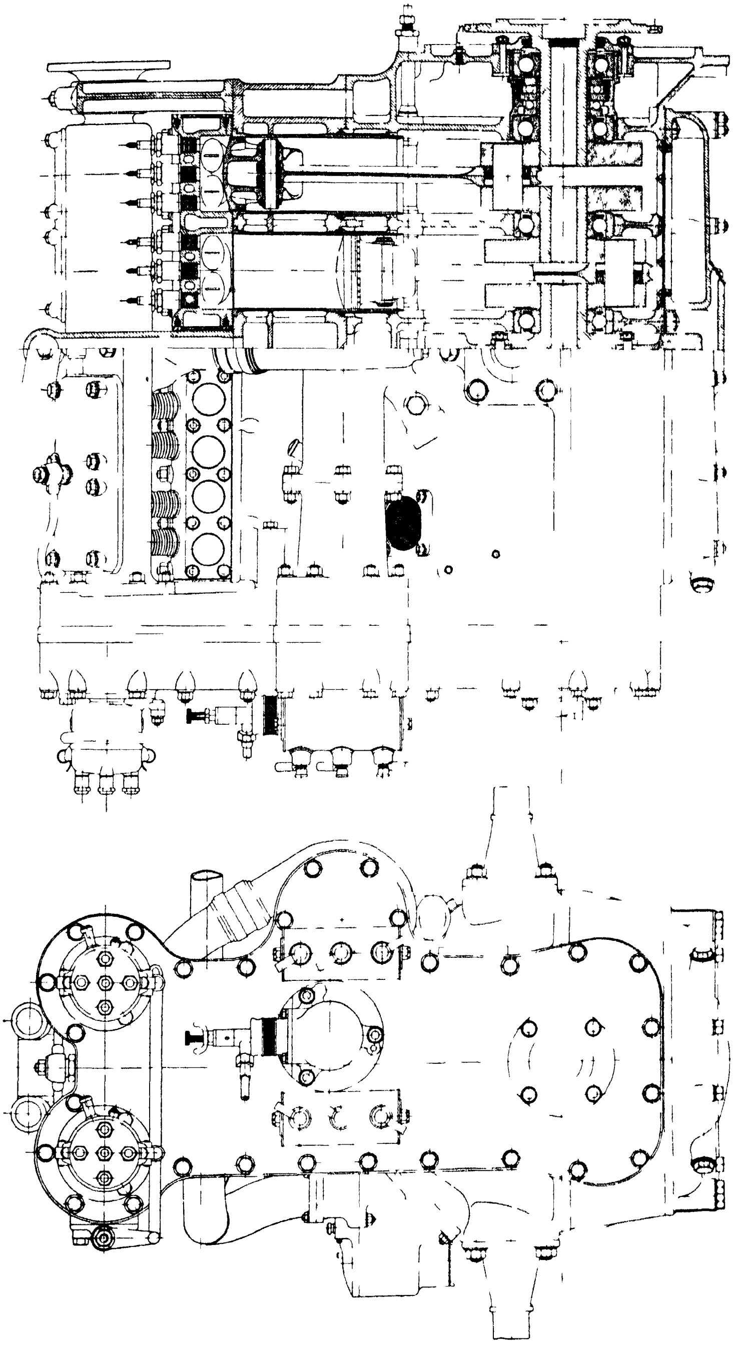


Fig 182—Vauxhall Three litre Racing Engine

distance from the sparking plug points to the farthest point in the combustion chamber is only 1·9 inches. In addition to the central plug, provision was made for the fitting of two other plugs, one on either side, to be operated synchronously from a single low-tension contact breaker. These additional plugs were intended rather as

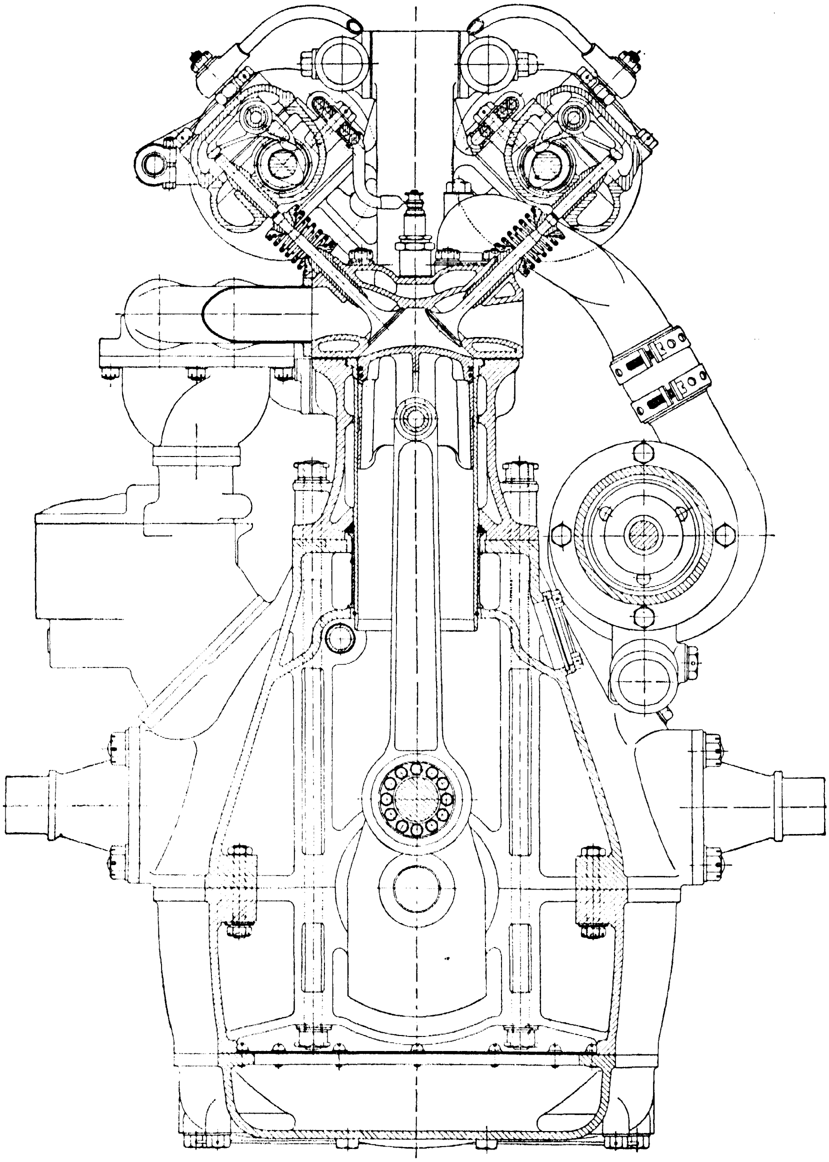


Fig. 183.—Vauxhall Three-litre Racing Engine

a standby in case of failure of the central plug; they were, in fact, never used.

(2) In order to ensure the maximum of structural rigidity the crankcase is made as deep as possible and of a barrel shape, with the maximum cross-section at the centre; the cylinder water-jackets

Fig. 184 —Vauxhall Three-litre Racing Engine: Auxiliary Drive

were cast in one piece and rigidly attached to form an additional girder, while through bolts extend from the cylinder block to the very bottom of the crankcase, thus forming a structure of extreme rigidity both as regards torsion and bending.

(3) With a view to eliminating torsional vibration, the flywheel is mounted in the centre of the crankshaft so that the maximum length subject to torsion is reduced to about 8 inches. The shaft is, in fact, made in the form of two entirely separate two-throw

cranks, each provided with flanges between which the flywheel is bolted. This arrangement, although very unorthodox, proved most successful, no trace of torsional vibration being observed at any speed at which the engine could be run.

(4) With a view to obtaining the highest possible mechanical efficiency, pistons of the slipper type are employed and the cylinder liners are maintained at a high temperature in order to lower the viscosity of the lubricant adhering to them. This latter is accomplished by isolating the lower part of the liners from the main water circulation, so that the cooling water surrounding them is left practically stagnant. For the rest, the use of ball and roller bearings wherever possible contributed to reducing the friction losses to the lowest possible limit and at the same time obviated the necessity for any form of oil cooling—always a troublesome problem.

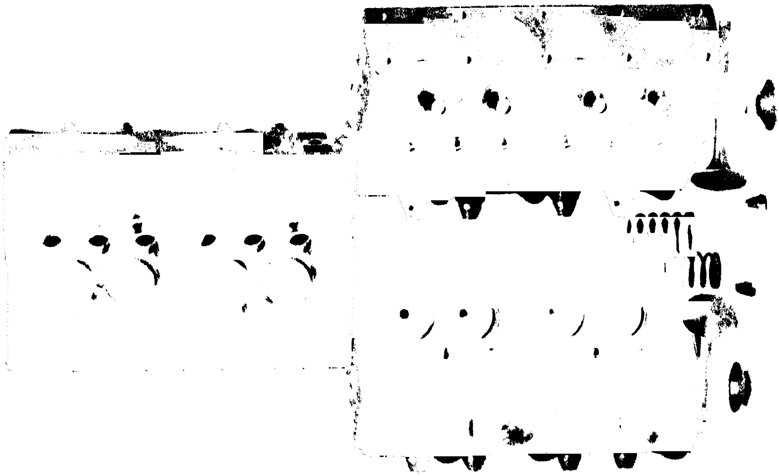


Fig 185 - Vauxhall Three-litre Racing Engine· Cylinder Heads and Valves

(5) In order to obtain a high volumetric efficiency the induction system was divided so as to avoid any overlapping of the suction strokes. The central pair of cylinders were fed from one carburettor and the outer pair from a second and entirely independent carburettor. In this manner use could be made of the kinetic energy of the gases flowing in the induction pipes thoroughly to fill the cylinders without any risk of one cylinder robbing another, as must inevitably occur when all four cylinders draw from any one induction manifold, owing to the overlapping of the opening period of the inlet valves.

The engine has four cylinders, each of 85 mm. bore and 132 mm. stroke, and was designed with a view to running continuously at from 4000 to 4500 R.P.M. with short periods up to 5000 R.P.M. In order

Fig. 186.—Vauxhall Three-litre Racing Engine: Crankcase

both to provide structural rigidity and to enclose the central flywheel the crank chamber is of barrel shape and is mounted in the chassis by trunnions attached to the sides. The cylinder block consists of an aluminium casting forming the water-jacket into which loose steel

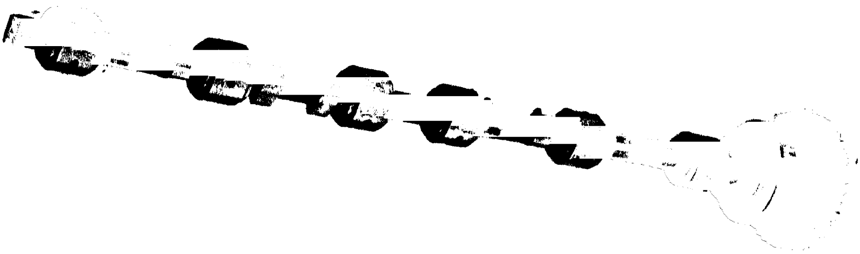


Fig. 187.—Vauxhall Three-litre Racing Engine: one of the Camshafts

liners are fitted, with rubber rings to ensure water-tightness. The cylinder heads are cast in pairs in hard bronze and call for no particular comment. The valves, of which there are four in number to each cylinder, are comparatively small with a high lift, and the valve gear generally is designed to operate at a speed of 5000 R.P.M.

The inlet valves are heavily masked, and by this means it is possible to employ a comparatively low rate of acceleration and therefore to use very light and lightly stressed valve springs. The two camshafts are carried in aluminium housings supported from the main cylinder block at their centre and at either end. They run in plain bearings with cast-iron floating bushes. The cams themselves are of very small diameter in order to reduce to the minimum the rubbing velocity; they are of the plain tangent flank form. The cam followers are of the plain curved slipper type, with a short straight push rod interposed between the follower and the valve itself. The camshafts are driven by means of a chain of spur gears, the intermediate pinions of which are carried in separate spider housings to permit of the meshing being correctly adjusted.

(6) Experience with bearings, both in actual engines and under separate tests, had shown that, even under the most favourable conditions as to lubrication, plain white-metal lined bearings could not be relied upon for the connecting-rod big-ends, because no matter how much oil be circulated through the bearings, nor how thoroughly it be cooled, there was little hope of getting rid of the heat generated by friction at a rate sufficient to keep the temperature of the bearing material within safe limits. Further, in order to reduce vibration and ensure structural rigidity, it was essential to keep the cylinder centres as close together as possible, and this limited the permissible width of bearing surface both on the crankpins and main crankshaft journals.

In view of these considerations it was apparent that if a continuous mean speed of over 4000 R.P.M. was to be maintained some form of crankpin bearing other than a plain white-metal lining would have to be employed. The choice lay between (*a*) the use of a floating bush between the crankpin and connecting-rod, under forced lubrication, and (*b*) a roller bearing. Both these would necessitate the use of some form of built-up crankshaft with case-hardened crankpins, since neither the floating bush nor the roller race could be split, nor could the weight of a split big-end bearing be tolerated. Of the two alternatives the floating bush would require continuous lubrication under pressure, while the roller bearing could be used with splash lubrication. Since for the same and other additional reasons, it was essential to employ ball or roller bearings for the main journals, the provision of continuous lubrication under pressure to the crankpins became a very difficult problem, and it was decided therefore to adopt the second alternative and employ roller bearings. The method of building up the crankshaft was another problem, and after much consideration it was decided to employ a completely built-up crank, consisting of plain parallel pins on to which the crank-cheeks were shrunk as in marine and large gas-engine practice.

The crankshafts throughout were made from plain mild steel with the pins case-hardened. For the connecting-rod big-end bearing, it was decided to use a double row of short rollers located in a one-piece bronze cage, while the hardened eye of the connecting-rod itself formed the outer race. This is further stiffened by means of two circular webs. Like the crankshafts, the connecting-rods are of plain low-carbon case-hardened steel.

Lubrication.—Two oil pumps of the oscillating valveless plunger type are provided, both of which are operated from one of the idle wheels of the gear train. One pump draws oil from the oil sump and delivers it to an oil gallery running the full length of the crankcase, provided with four jets playing oil on to each of the crank-throws. The second pump delivers oil under a pressure of about 25 lb. per square inch to the camshafts, the oil being distributed through the hollow fulcrum pins of the valve rockers; from the camshaft casings, the oil drains back by gravity to the crankcase.

Cooling.—The cooling water is circulated by means of a centrifugal pump running at one-half engine speed. From the pump the water passes around the upper end of the cylinder liners, the lower parts of which are partitioned off in order to maintain the water more or less stagnant, and so to permit of the bearing portion of the liners attaining rapidly a fairly high temperature. From the upper deck of the cylinder block the water passes to the cylinder heads in parallel, and so returns to the radiator. In order both to cool the crankcase and slightly to heat the air on its way to the carburettors, the air supply to the engine is drawn through the upper part of the crankcase and around the exposed lower ends of the cylinder liners.

The compression ratio used is 5·8 : 1. It had originally been proposed to employ a much higher compression ratio, and to run on a special fuel mixture, but owing to the difficulty of providing an efficient form of combustion chamber with any higher compression ratio, and at the same time avoid any risk of the pistons striking the valves should these accidentally stick in the full open position, it was considered safer to employ a lower ratio, at which, owing to the short-flame travel from the sparking plug, ordinary good quality petrol can be used without detonation.

In general, though the engine is designed throughout to run at very high speeds, it contains no extremes either of design or material. The whole of the crankshaft, the connecting-rods, and the gudgeon pins are of straight low-carbon mild steel. Neither the connecting-rods nor the pistons are particularly light. The cams are of the plain tangent flank form, free from any concave surfaces, the acceleration of the valve gear is low, and the valve springs are very lightly stressed. In short, the engines were designed throughout with a view to reliability both in manufacture and in running, and an ample margin of safety provided for.

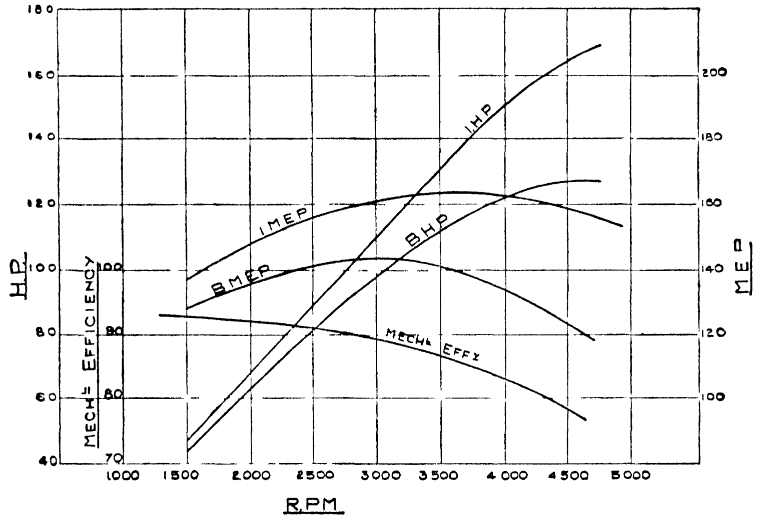


Fig. 188 —Vauxhall Three-litre Racing Engine

The performance of one of these engines which underwent prolonged testing on the test-bench is shown in fig. 188, from which it will be seen that a maximum of 129 B.H.P. is reached at a speed of 4500 R.P.M., the brake and indicated mean pressures at this speed being 124 and 159 lb. per square inch respectively, and the mechanical efficiency 78 per cent. It will be observed also that the highest indicated mean pressure was obtained at a speed of about 3700 R.P.M., showing that the combination of induction pipe design and valve setting was such as to give maximum over-all efficiency at this speed. With a combustion efficiency of 34·75 per cent this corresponds to a volumetric efficiency at N.T.P. of 80·3 per cent, a figure which is in very close agreement with that obtained from the author's variable compression engine under similar conditions as to temperature, and at a speed of 1750 R.P.M., which is the most efficient speed for this particular engine. This figure also is in very close agreement with readings taken of the compression pressure when motoring, which at 4000 R.P.M was found to be 139 lb. per square inch, indicating that the cylinders were filled up to very nearly full atmospheric pressure at this speed. The mechanical efficiency was arrived at from a very large number of tests by Morse's method on pairs of cylinders separately, on individual cylinders, and by motoring tests. The three methods showed exceptionally close agreement over the whole range of speed.

The performance of the engines is almost exactly what might be anticipated from an analysis of the general design based on the data given in the preceding chapters, and, since it conforms so closely, it may be of interest and perhaps of some use to engine designers to recapitulate such data in so far as it applies to these particular engines. It may also, the author hopes, help to dispel the still prevalent superstition that some mystery enshrouds the performance of racing motor-car engines, whose behaviour is, in fact, perfectly normal in every respect.

The leading dimensions of these engines are as follows:—

Bore	85 mm. = 3·31 inches.
Stroke	132 mm. = 5·2 ,,
Compression	5·8 : 1.
Area of piston	8·75 square inches.
Swept volume of cylinder	45·5 cubic inches.
Weight of reciprocating parts (per cylinder)	1·7 lb.
,, ,, ,, (per sq. in. of piston area)	0·195 lb. per sq. in.
Number of valves	4– 2 inlet, 2 exhaust.
Diameter of valve ports (inlet)	- 1·31 inches.
,, ,, (exhaust)	= 1·30 ,,
Lift of all valves	0·354 ,,
Effective area through inlet valves	2·55 sq. in.
(Inlet valves masked 0·050 inches of travel.)	
Ratio piston area to effective inlet port area	- 3·44 : 1.

Curve A in fig. 189 indicates the mean gas velocity through the inlet valves at speeds varying up to 4500 R.P.M., and curve B shows the volumetric efficiency which corresponds with these mean gas velocities allowing for the latent heat of evaporation of the fuel, assuming that the minimum of preheating were used and that there were no undue wiredrawing in the induction pipe or carburettors, both of which conditions apply in this case. Curve B is arrived at by calculations such as those given in Chapter II, by deduction from various test results, and finally by direct air measurements taken on various engines with similar combustion chambers and similar gas velocities through the inlet valves. The falling off in volumetric efficiency at the lower speeds is due merely to the late closing of the inlet valves, which results in the rejection of some of the combustible mixture during the early part of the compression stroke; while the falling off again at very high speeds is due to wiredrawing or insufficient valve area. With a combustion chamber of the form used giving the maximum of turbulence, and with central ignition, the efficiency reckoned on the air consumption will, at high speeds, be approximately 69 per cent of the air-cycle efficiency for the compression ratio (*vide* Chapter IV).

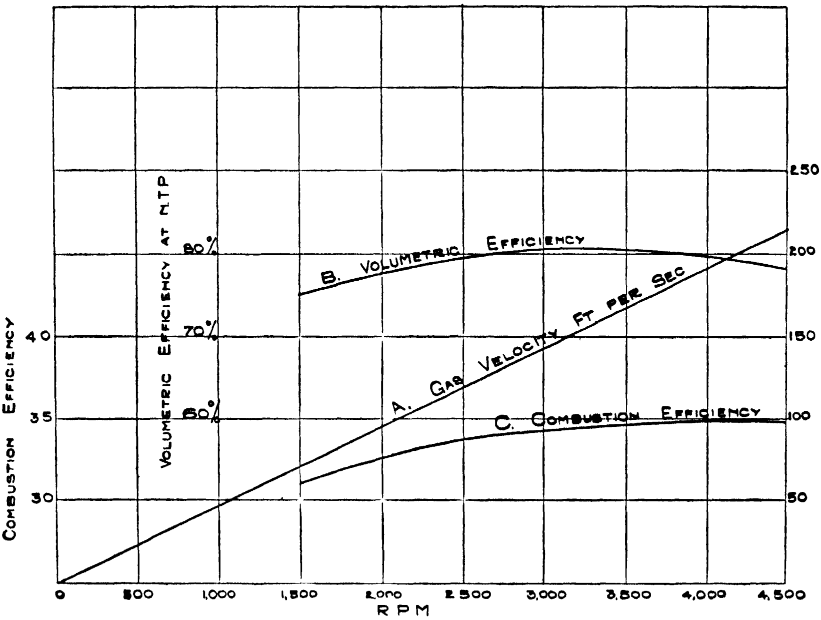


Fig. 189.—Vauxhall Three-litre Racing Engine

The air-cycle efficiency corresponding to a compression ratio of 5·8 : 1 is 50·5 per cent, so that the combustion efficiency will be $\frac{69}{100} \times \frac{50 \cdot 5}{100} = 34 \cdot 8$ per cent at the highest speeds, when turbulence is at a maximum and direct heat loss at a minimum. As the speed is reduced turbulence becomes less, owing to the lower entering gas velocity, and the direct heat loss, though of comparatively small consequence, will also increase, with the result that combustion efficiency may be expected to vary approximately as the curve C, fig. 189. The values given in this curve are taken from those found on the author's variable compression engine under almost exactly similar conditions.

From the curves B and C and the known heat of combustion per standard cubic inch of mixture, as given in Chapters I and II, the indicated mean effective pressure can be arrived at directly by multiplying together the volumetric efficiency, the heat of combustion per standard cubic inch (47·8 ft.-lb. in this instance) × 12 × the combustion efficiency; thus at 3000 R.P.M. the indicated mean pressure will be $0{\cdot}807 \times 47{\cdot}8 \times 12 \times 0{\cdot}343 = 159$ lb. per square inch. Similarly at speeds from 1500 to 4500 we find that the theoretical indicated mean effective pressure should be—

R.P.M.	Volumetric Efficiency per cent.	Combustion Efficiency per cent.	Indicated Mean Pressure lb. per sq in.	Observed I.M.E.P. lb. per sq. in.
1500	75·0	31·0	133·4	138·5
2000	77·6	32·6	145·1	148·0
2500	79·7	33·9	155·0	156·0
3000	80·7	34·3	159·0	161·0
3500	80·7	34·6	160·4	162·5
4000	79·9	34·7	159·2	162·2
4500	78·5	34·8	156·9	159·0
5000	77·0	34·8	153·7	..

From the above it will be seen that the agreement between the estimated and the observed figures is so close as to prove that the engine is behaving normally in every respect; it is, in fact, so close as to indicate a certain share of coincidence, for neither calculation nor measurement could be exact to within closer limits than 1 per cent, leaving scope for a variation of 3 lb. per square inch between the observed and the calculated figures, assuming that all the premises on which the latter were based were strictly accurate. No tests were made above 4500 R.P.M. or below 1500.

The losses in any internal-combustion engine may be divided up into—(1) Piston friction. (2) Fluid pumping losses. (3) Bearing friction and auxiliary drives.

Of these piston friction constitutes always by far the largest proportion, and all more recent tests appear to indicate that for a piston of more or less normal design and proportions, and for normal conditions as to lubrication and jacket temperature, the piston friction in terms of lb. per square inch on the piston head may be arrived at with a fair degree of approximation by the empirical formula—

$$\left\{\frac{\frac{\text{Mean fluid pressure including compression}}{4}+\frac{\text{2 Mean inertia pressure}}{3}}{10}\right\}+2\cdot 0.$$

From such a formula we find that the piston friction expressed in terms of mean pressure on the piston at every fourth stroke (i.e. expressed on equal terms to the useful mean pressure) should be—

Speed R.P.M.	Piston Friction (lb. per sq. in. M.E.P.)
1500	7·7
2000	8·6
2500	10·0
3000	11·4
3500	13·1
4000	15·1
4500	17·3

The fluid pumping losses are dependent, assuming a normal valve setting, as in this case, and no other serious obstruction to the flow of gas into or out of the cylinders, upon the mean gas velocity through the valves and more particularly through the inlet valves. Fig. 190 shows the observed fluid pumping losses at different gas velocities: from this curve, and that of gas velocity given in fig. 189, we find that the fluid pumping losses in this particular engine again expressed in terms of mean pressure should amount to—

Speed R.P.M.	Fluid Pumping Losses (lb. per sq. in.)
1500	2·0
2000	2·3
2500	2·9
3000	4·0
3500	5·4
4000	6·9
4500	8·6

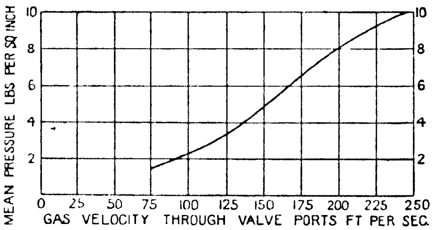


Fig. 190.—Mean Pressure of Suction Loop for different Gas Velocities with Normal Valve Setting and Poppet Valves

Finally, there remains the friction of the bearings and the power absorbed by the auxiliary drives: these latter consist of a long train of gears to operate the overhead camshafts, the camshafts themselves in plain floating bearings, a large water circulating pump, two oil pumps, a small air-compressor for fuel supply, and the ignition gear. No direct measurement was taken of the power absorbed by these auxiliaries, and in the absence of actual data we must fall back on analogy from tests on other engines more or less similarly equipped. From such an analogy it may be assumed that

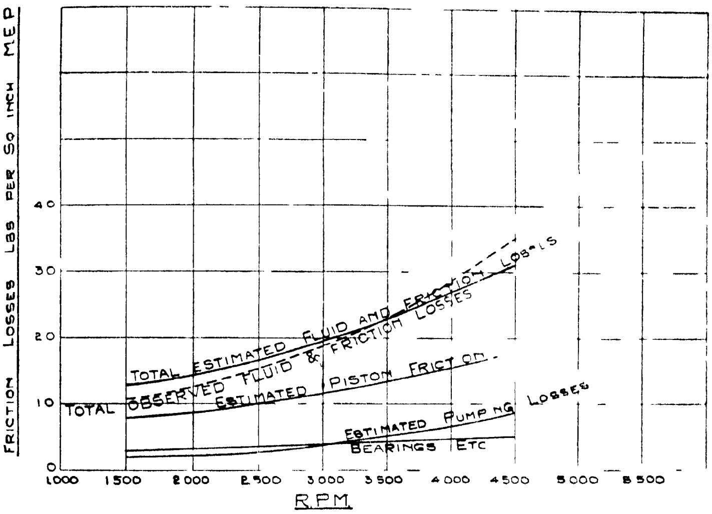


Fig 191 —Estimated and observed Fluid and Friction Losses

the loss due to all these sources will range in more or less a straight line from the equivalent of 3 lb. per square inch at 1500 R.P.M. to about 5 lb. at 4500 R P.M.

From the above it will be seen that the total of fluid and frictional losses may be estimated at—

Speed R P.M.	Total Losses (lb per sq in M E P) estimated	Observed Losses
1500	12·7	10 9
2000	14 2	12·6
2500	16·5	15·2
3000	19·3	18·6
3500	22·7	22 7
4000	26·6	28·8
4500	30·9	35·0

Fig. 191 shows in full lines the estimated friction losses as arrived at in the above tables, and in dotted line, the observed losses as arrived at by motoring and by tests with the ignition cut off from various cylinders. It will be observed that while the general slope of the curve of mechanical and other losses does not agree very closely with the estimated curve, yet the mean value throughout the whole range of speed is in very fair agreement.

A number of readings of fuel consumption using the petrol referred to—as sample A in Chapter I—were taken at different speeds and loads, using in all cases an economical carburettor setting: that is to say, about 10 per cent weak as against the 20 per cent rich mixture employed for the attainment of the utmost possible power output. The fuel consumption readings therefore were taken with the carburettor so adjusted as to reduce the maximum power by about 6 per cent. These tests gave the results shown in figs. 192 and 193, in terms of pints per hour per indicated and per brake horsepower. This particular petrol has a corrected calorific value (including the latent heat of evaporation of the liquid) of 18150 B.T.U.s per pint.

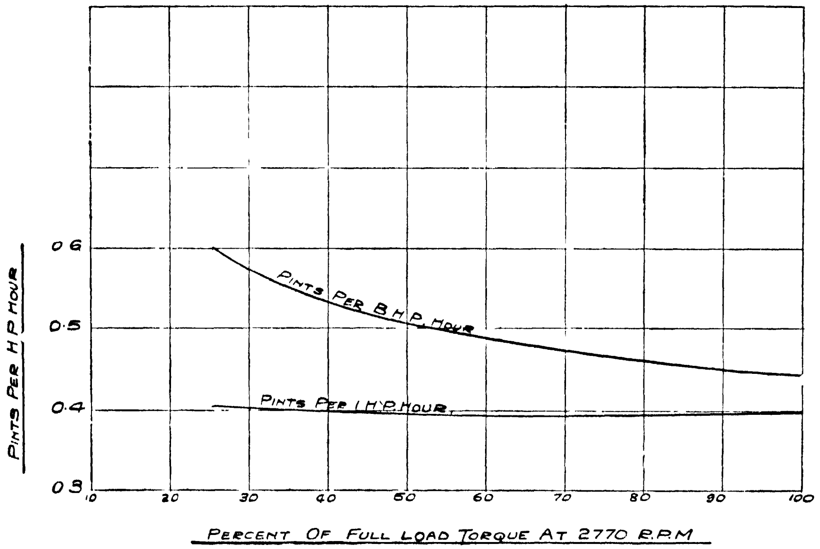


Fig. 192.—Three-litre Vauxhall Racing Engine

Curve showing Consumption on Throttle at 2770 R.P.M., and most Economical Mixture Strength. Fuel, Petrol. Oil, "Shell", L.R.O.

It will be seen that at 3000 R.P.M. the fuel consumption on full load is only 0·45 pint per B.H.P. and 0·395 pint per I.H.P. hour, corresponding to a brake thermal efficiency of 31·2 per cent and an indicated thermal efficiency of 35·4 per cent, a figure actually slightly in excess of the computed combustion efficiency, while at 66 per cent full load torque the observed fuel consumption at about this speed was only 0·48 pint per B.H.P. hour, a figure unequalled even at full load by any touring-car engine. These figures should go far to dispel the theory that a racing-car engine is essentially extravagant in fuel. The readings of fuel consumption when the load was reduced by throttling down to one-third full load torque are rather striking.

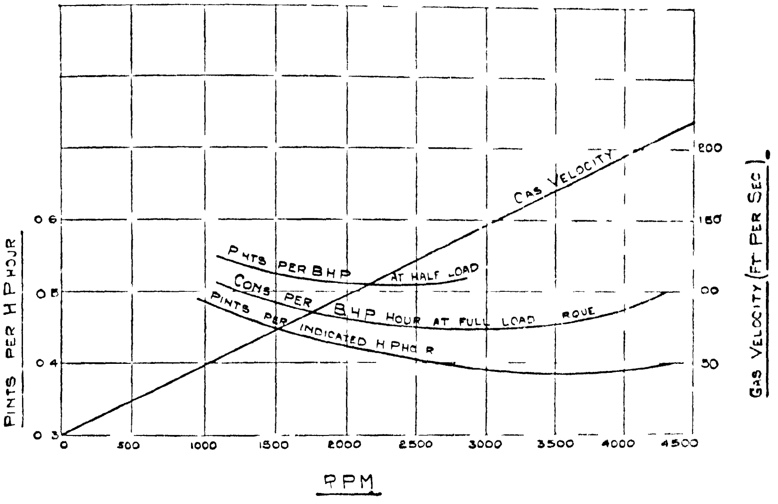


Fig 193.—Three litre Vauxhall Racing Engine

Curve showing Variation in Fuel Consumption with Speed Most Economical Mixture in all cases. Fuel, Petrol. Oil, "Shell", L R.O.

Thus at 2770 R.P.M. the fuel consumption on petrol was found to be—

B.H.P.	Fuel Consumption (Pints per B H P. hour)
80	0·455
70	0·465
60	0·480
50	0·505
40	0·535
30	0·565
20	0·610

At 1950 R.P.M. the fuel consumption was—

B.H.P.	Fuel Consumption (Pints per B.H.P. hour)
60	0·470
50	0·480
40	0·495
30	0·525
20	0·570
15	0·605

It will be noted that when developing 20 B.H.P. at 1950 the gross consumption is only $20 \times 0{\cdot}570 = 11{\cdot}4$ pints per hour. With the gear ratio used 1950 R.P.M. corresponds to a road speed of 48·5 M.P.H., a speed which calls for an expenditure of just about 20 B.H.P. reckoned at the engine shaft, so that the consumption in miles per gallon with an economical carburettor setting even at this high mean speed should be $48{\cdot}5 \times \frac{8}{11{\cdot}4}$ = about 33·9 miles per gallon.

A brake thermal efficiency of 31·2 per cent is, the author believes, the highest ever yet achieved by any engine running on petrol. Incidentally it is worthy of note that the indicated thermal efficiency reckoned from the fuel consumption, when using a weak mixture, corresponds very closely with the calculated combustion efficiency, showing that there can be practically no loss of unburnt fuel by irregular distribution or indeed from any other cause. It is interesting also to note that the fuel consumption per I.H.P. hour is exactly the same as that obtained in the single cylinder variable compression engine (described in Chapter I) when running on the same fuel, at the same compression ratio, and at the same gas velocity through the valves, but the brake thermal efficiency of the racing engine is considerably greater owing to the higher mechanical efficiency of the latter.

The author has dealt at considerable length with this particular engine, because the results obtained from it serve admirably to emphasize that a racing-car engine is nothing more or less than a highly efficient internal-combustion engine designed throughout on purely scientific lines, whose behaviour is, from a thermodynamic point of view, perfectly normal in every respect.

Fig. 194 shows a photograph of one of the racing-cars fitted with the engine described above. Fully equipped, as shown in the photograph, the car weighs with driver and mechanic 2700 lb.; in this condition and with a gear ratio giving 25 miles per hour at 1000

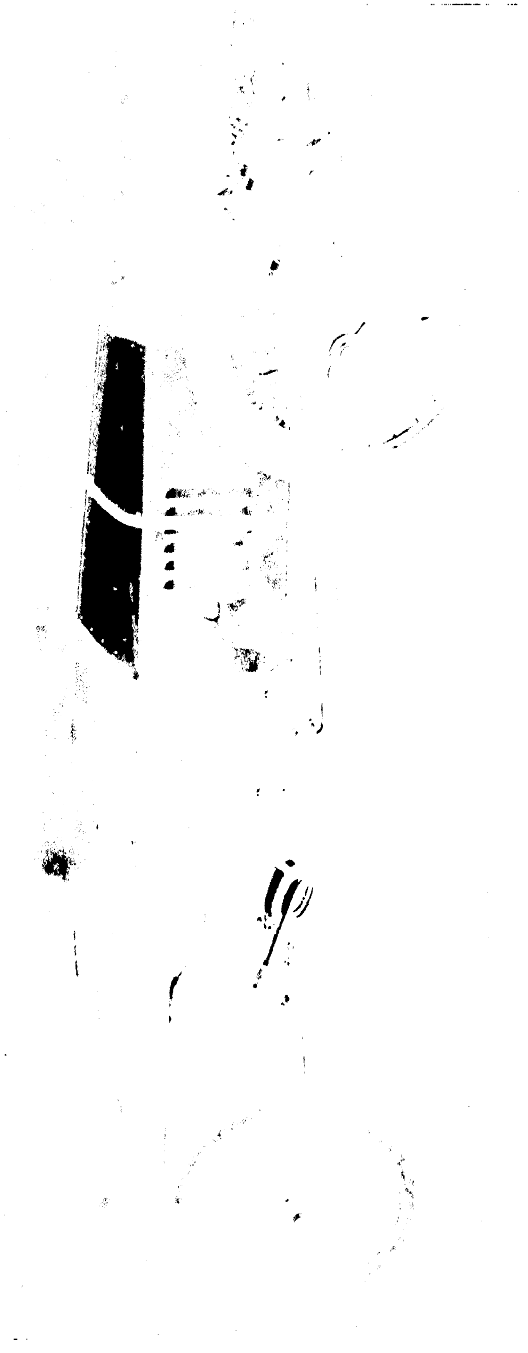


Fig 194.—Vauxhall Three litre Racing-car

R.P.M. it is capable of a maximum speed on the level of 115 miles per hour, corresponding to an engine speed exclusive of wheel slip of 4600 R.P.M., the actual engine speed being probably about 4800 R.P.M.

It is interesting to note that one of these cars after nearly eight years' use, and when fitted with a super-charger, won a number of competitions against much more modern racing-cars, and in 1929 established a new record for the annual Shelsley Walsh hill climb for cars of unlimited size or power.

CHAPTER XI

HIGH-SPEED HEAVY-DUTY ENGINES FOR TANKS

Although the conditions applying to an engine for tanks are somewhat specialized owing to the peculiar nature of the service required of them, yet, apart from certain features, the following examples may be taken as fairly typical of large high-speed heavy-duty engines developed during the war of 1914–18. Unlike most other heavy transport duties, the engines for tanks were called upon to run for comparatively long periods under very heavy loads, the average load factor when travelling across rough country being over 80 per ~~cent as compared~~ with the 35–45 per cent load factor of ordinary motor lorry engines; again, the engines ran always at their governed speed, which ranged from 1200 to 1350 R.P.M. and averaged about 1250 R.P.M., corresponding to a normal piston speed of 1560 ft. per minute, or about double the average piston speed of motor lorry engines.

Owing to the very large amount of dust and mud imported into the tank by the creeping tracks, the engine was always smothered in dirt or dust, and for this reason it was very desirable totally to enclose the crankcase and to eliminate breathers or any other form of ventilation. Further, they were required to use inferior fuel, and in many cases received only the most scant and unskilled attention.

Owing to the severe gradients which the tanks were capable of negotiating, the engine was frequently required to operate at an angle of over 35° to the horizontal, as shown in the photographs, figs. 195 and 196, which show a tank climbing out of a deep trench, while fig. 197 shows some of the other duties expected of a tank. Further, it was laid down by the authorities that under no circumstances should the engines show smoke from the exhaust. These two conditions necessitated the adoption of special measures both as regards the lubrication and the piston design.

By reason of the low priority under which tanks and their equipment were constructed until the very last phase of the war, only the cheapest and most easily worked materials could be used. The allowance of aluminium available was so small that it sufficed only for the pistons and induction pipes, while the use of high tensile steel was entirely banned.

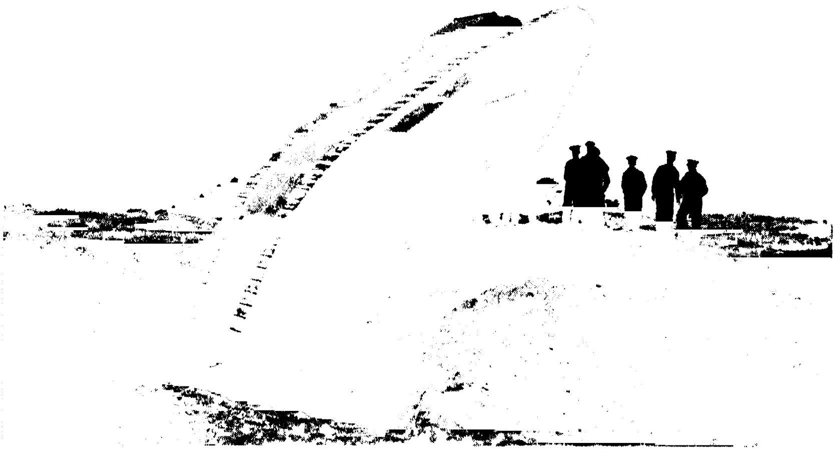


Fig. 195.—Tank at about Maximum Climbing Angle

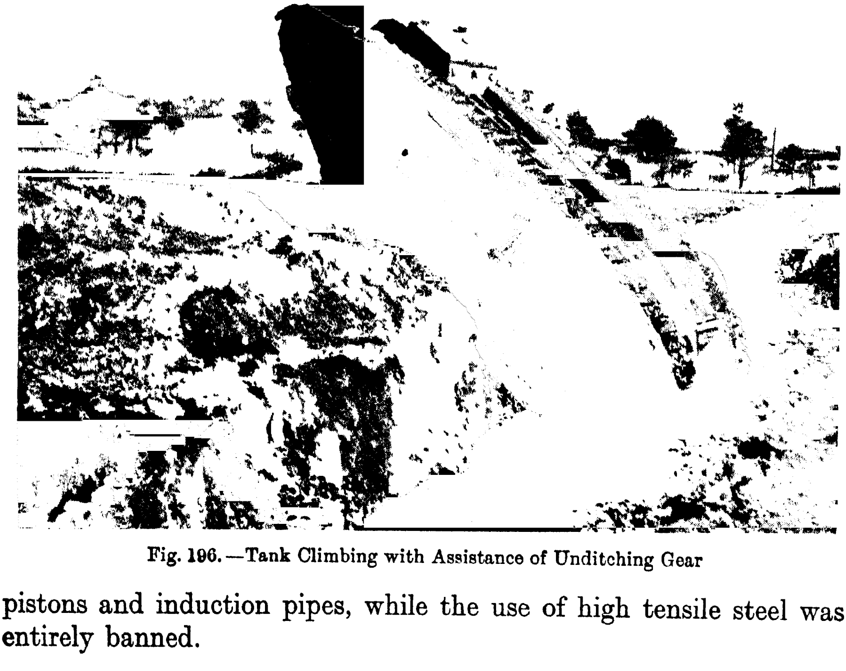


Fig. 196.—Tank Climbing with Assistance of Unditching Gear

The standard 150 H.P. type is shown in the photos, figs. 198–201, and 216, and in the drawings, fig. 202. Six separate cylinders were employed, each of $5\frac{5}{8}$-in. bore and $7\frac{1}{2}$-in. stroke; the water jackets were arranged with large openings at the sides, which were covered

Fig. 197.—Tank crossing wide River

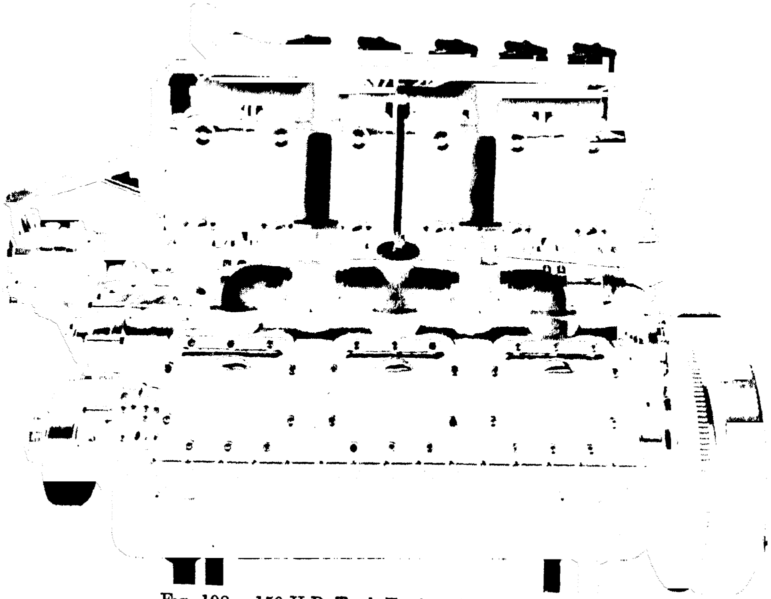


Fig. 198.—150 H.P. Tank Engine, Carburettor Side

with screwed-on sheet steel doors. This form of construction, in addition to facilitating the foundry work, allowed of the cylinder centres being brought very close together, thus reducing both the over-all length, which was very limited, and the bending moment on the crankcase due to the two opposing couples formed by each group of three pistons.

The cooling water was delivered to the bottom of the water jacket on the side remote from the valves, and the outlet was arranged between the two sparking plug bosses on the opposite side of the cylinder, the object being to ensure a rapid circulation of water round the sparking plugs.

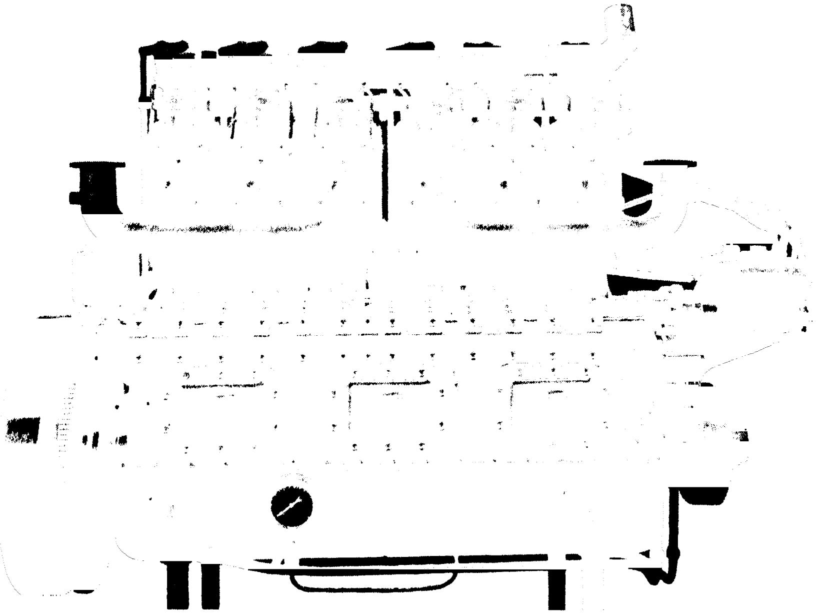


Fig. 199.—150 H.P. Tank Engine, Exhaust Side

Provision was made in the cylinder heads for fitting compressed-air starting valves, although this system of starting the engines was never employed.

So far as the exhaust valves were concerned, there is nothing very special to record. Care was taken to ensure the best possible cooling of these by providing a wide seating with an ample supply of water all round, and by using a valve stem of large diameter to conduct the heat away. The valve was cooled by carrying the water as close as possible up to the head of the valve, and also by the use of a valve guide of phosphor bronze, which is an excellent conductor of heat. There is one feature, however, in connection with the exhaust valves which perhaps calls for comment—that is, they were of 3 per cent nickel steel, case-hardened all over. The object of this treatment was twofold:—

(1) Although, of course, the head of the valve does not remain hard, the carbonized surface resists pitting, with the result that the seating lasts much longer. Grinding-in was therefore seldom necessary.

Fig. 200.—150 H.P. Tank Engine, Flywheel End

(2) The case-hardened stem rendered possible the use of a phosphor-bronze valve guide without risk of tearing or seizing.

The connecting-rods were mild-steel stampings of normal design. The only point for comment is in the length of the rods, which were 16 in. between centres, giving an l/r ratio of 4·26 : 1. The principal reason for the employment of these long rods lay in the fact that it was anticipated that a four-cylinder unit of this engine would be required at a later date, as indeed proved to be the case, and the shorter rods which it would have been possible to employ on a six-cylinder engine would have been a great disadvantage in a four-cylinder engine, on account of the secondary disturbing forces.

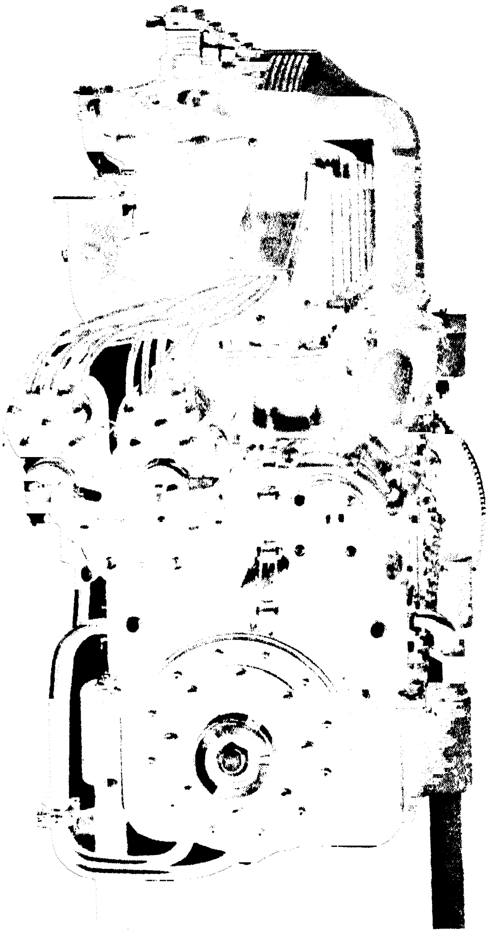


Fig. 201.—150 H.P. Tank Engine, Magneto End

The top half of the crankcase, or column, was an iron casting of an average thickness of $\frac{3}{8}$ in. The general design is clearly shown in the general arrangement drawings, fig. 202, and the function of the false top to the crankcase has already been explained in the chapter dealing with piston design, &c. Inspection doors were fitted

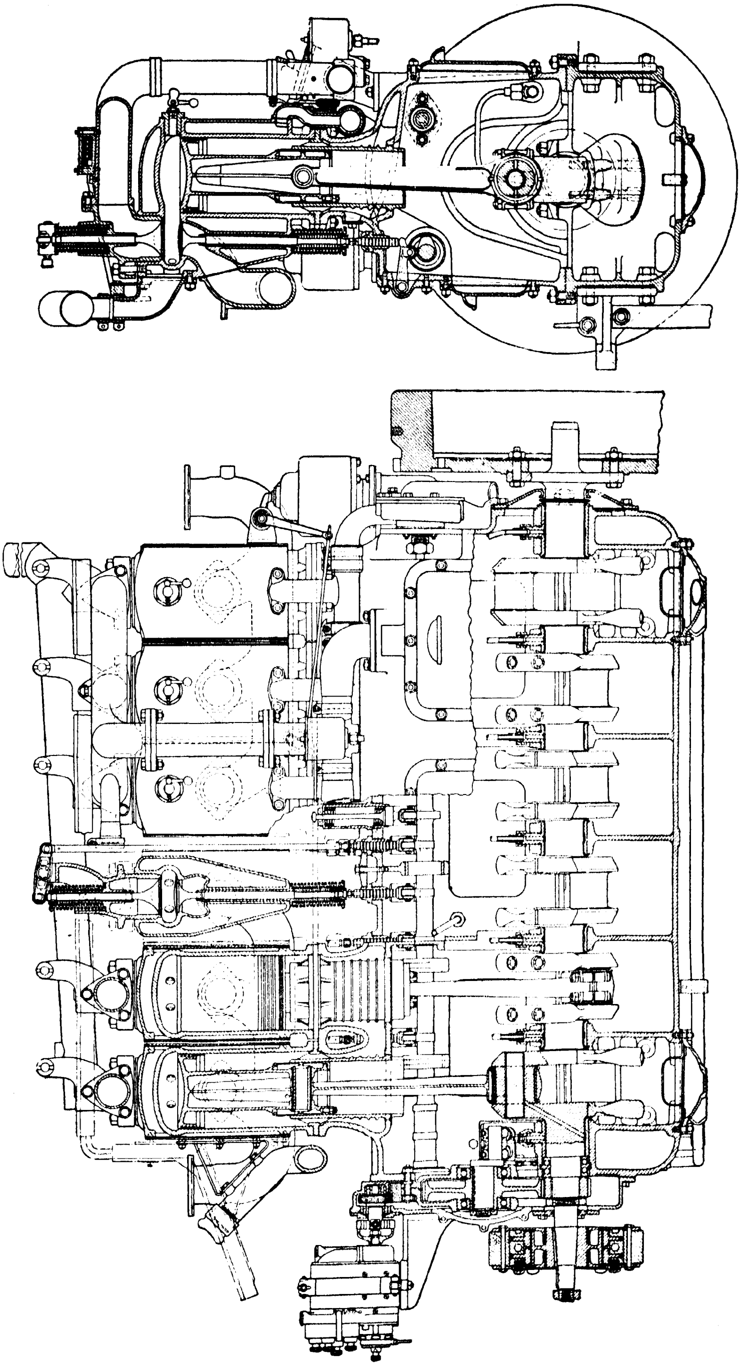


Fig. 202.—General Arrangement of 150 H.P. Engine, Longitudinal and Cross Sections

on both sides of the column, and the construction was such that it was possible to remove the connecting-rods, &c., through the inspection doors (fig. 203).

The crankshaft was mounted on seven plain bearings carried in the cast-iron bed-plate; the bearing caps were mild-steel stampings; and the white-metal-lined " brasses " were located in the bearing caps in order to allow of the removal of both halves of the journal bearings, should this be found necessary, without disturbing either the bed-plate or the crankshaft. Fig. 204 shows the arrangement of the lubrication connections to the journal bearings, and it will be seen that the oil pipe was attached directly to an extension piece cast integrally with the top half of the bearing brass. The extension piece passed through a hole drilled in the steel bearing cap, thus serving to locate the bearing shell. This method of construction

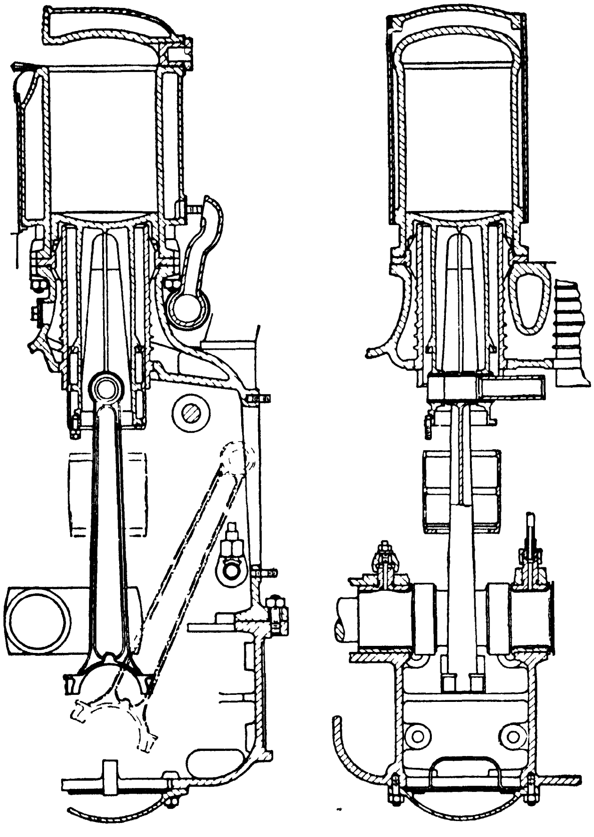


Fig. 203.—Connecting-rod, Dismantling Diagram

had the advantage that there was less tendency for oil to leak round between the bearing shell and its housing, and so insulate the brasses. The bearings thus disposed of their heat the more readily.

The crankshaft was a mild-steel forging, the principal dimensions of which are given in the table at the end of this description. Owing to the restrictions as regards the length of the engine, the available area of bearing surface was severely limited, and the difficulty of providing adequate bearing areas was still further increased by its being necessary to employ material for the crankshaft of very low surface hardness and having therefore very poor wearing properties. In apportioning the bearing surface between the connecting-rod and journal bearings in the original design, a higher load factor was allowed on the journal bearings and particularly the centre bearing, since this factor could be reduced, if found necessary, by the addition of balance weights.

The arrangement for the oil supply to the big ends was orthodox.

The arrangement of expanded-in tube in the crank-pin as shown did not prove altogether satisfactory in service, for it was found that there was a tendency for the annular space to become choked in course of time.

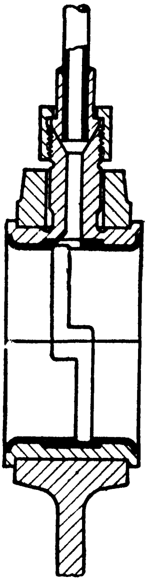


Fig. 204.—Crankshaft Journal Detail

The tube was therefore discarded and replaced by the usual arrangement of two end plugs retained in position by a single through bolt.

The flywheel was an iron casting 26 in. in diameter, and was bolted to a flange formed solid with the crankshaft. A Lanchester vibration damper was attached to the forward end of the crankshaft in order to damp out any torsional vibration.

In order to allow of the engine operating satisfactorily when tilted through large angles, the lubrication was on the " dry base " system; that is to say, the oil supply was not carried in the bedplate, but in a separate oil tank. Three oil pumps of the valveless plunger type were fitted, all three of which were driven from a single crank-pin, which in turn was driven by the intermediate timing gear wheel. The general arrangement of the oil pumps and their driving gear is shown in fig. 205. The centre pump circulated the oil through the various bearings, and the two scavenge pumps collected and returned the used oil to the external oil tank. Each of the scavenge pumps was connected up to one of the small oil sumps which were provided at each end of the bedplate. The lubrication pipework is shown in fig. 206, which also illustrates the oil relief valve at the flywheel end of the main oil lead. In the original design, the scavenge pump suction pipes were arranged externally on the grounds of accessibility of the pipe joints, but in the Mark V Tanks, in which these engines were principally used, the joints were not accessible when the engine was mounted in position. Later engines were therefore fitted with the suction pipes inside the bedplate.

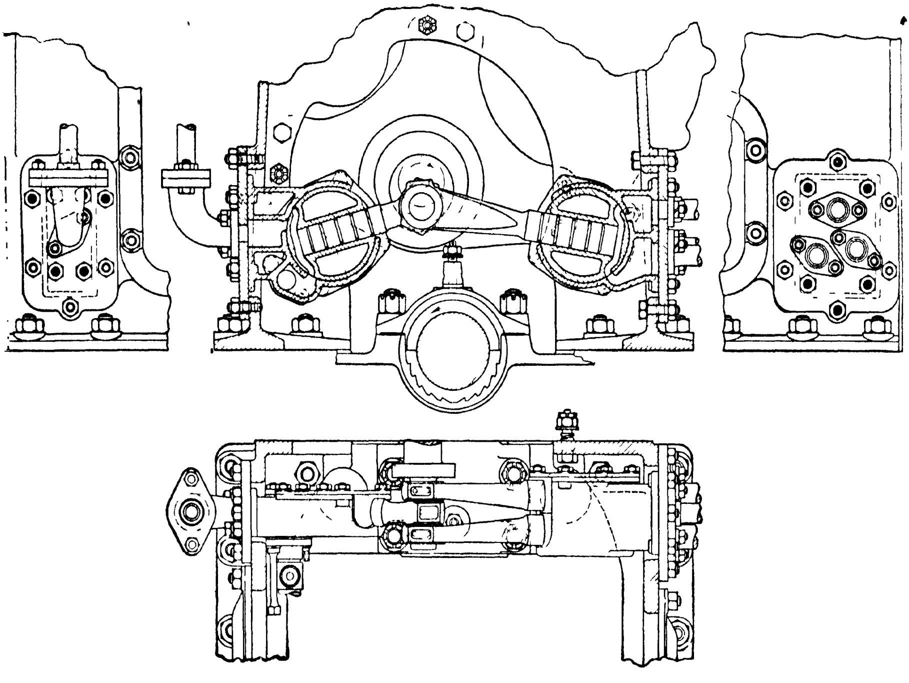


Fig. 205.—Oil Pump Arrangement

Owing to restriction in width, it was necessary to place all the auxiliaries at the ends of the engine. The auxiliaries to be provided for were as follows: Two magnetos, three oil pumps, two governors, water-circulating pump, and air-pressure pump. The arrangement of the various auxiliary drives will be seen from the illustration, and fig. 207 shows these diagrammatically. In the original design two governors were provided, one to limit the maximum speed of the engine, and the other to open the carburettor throttles directly the engine speed fell below 400 R.P.M. The object of the second governor was to prevent accidental stoppage of the engine. It was, however, found to be unnecessary, and only the high-speed governor was retained.

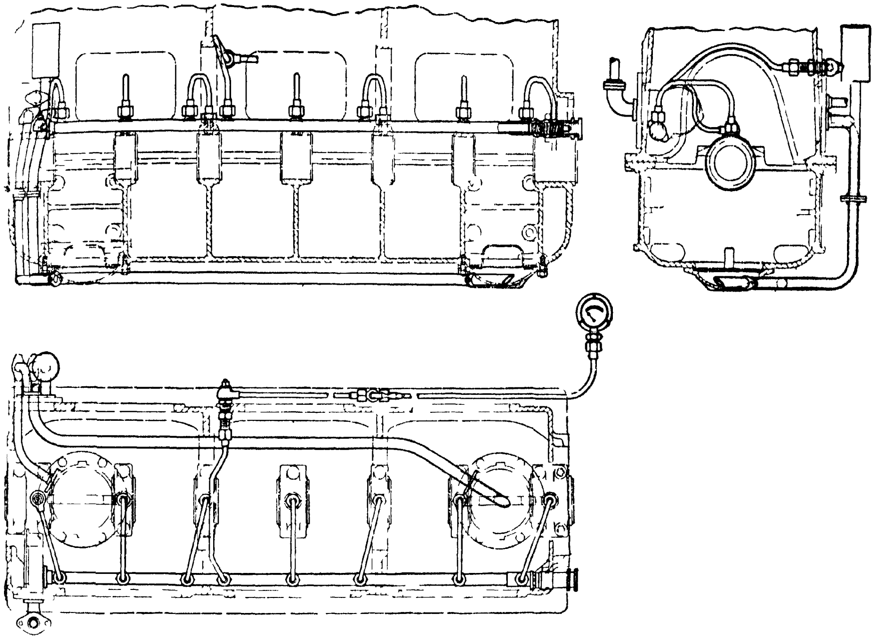


Fig. 206.—Oil Pipe Arrangement

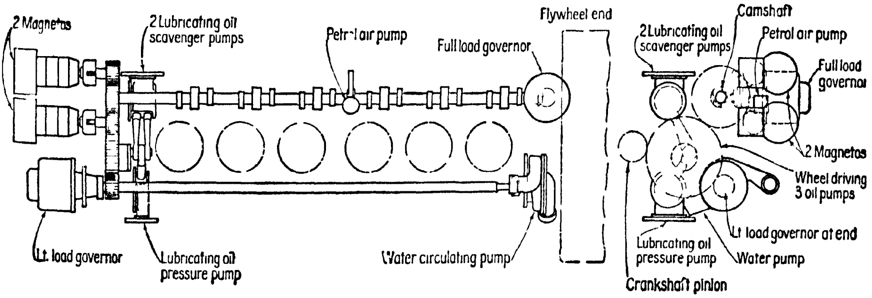


Fig. 207.—Auxiliary Drive Diagram

Fig. 208 shows the arrangement of the water-pump drive. The pumps were designed and made by the Pulsometer Engineering Co., and their performance is shown in the curve, fig. 209.

The intermediate timing gear wheel was mounted on ball bearings carried in a cast-iron spider bolted up to the front wall of the column, a form of construction which facilitates the correct meshing of the timing gear. All three oil pumps were driven by a small disc crank keyed to the hub of the intermediate wheel.

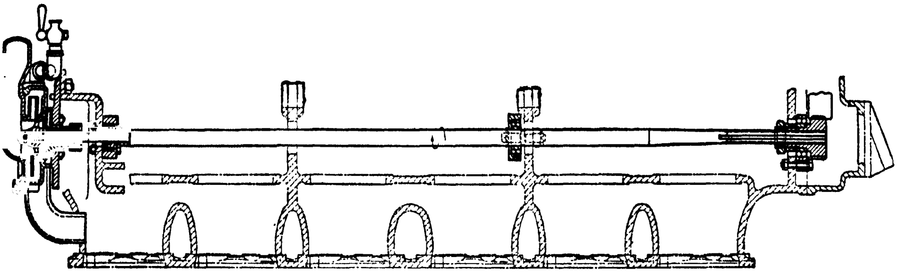


Fig. 208.—Water Pump Drive Arrangement

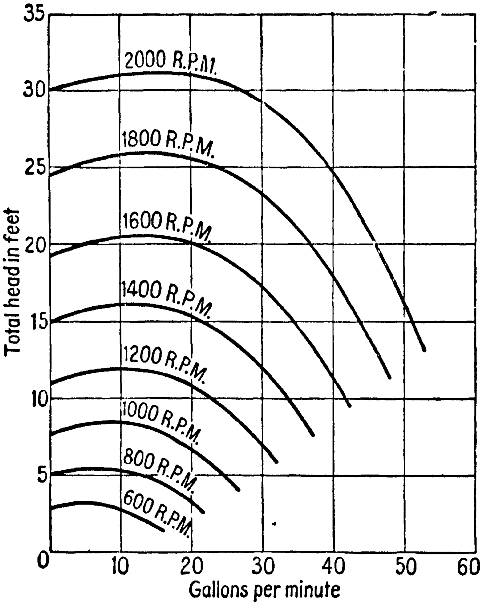


Fig. 209.—Performance Curves of Water Circulating Pump

Fig. 210 shows the general arrangement of the governor.

Two 55-mm. vertical Zenith carburettors were fitted, the whole of the air supply to which was taken from the chambers surrounding the cross-head guides; the method of warming the air supply to the carburettors has already been described in connection with the piston construction. A hand-adjusted cold-air valve was fitted between the two carburettors for use in very hot weather.

All engines were required to pass the following tests before acceptance:

(1) A full-load test of two hours' duration, during which the power must not fall below 150 B.H.P. at 1200 R.P.M. During this test the fuel and oil consumption was not to exceed 0·7 pint (petrol) and 0·02 pint (oil) per B.H.P. hour.

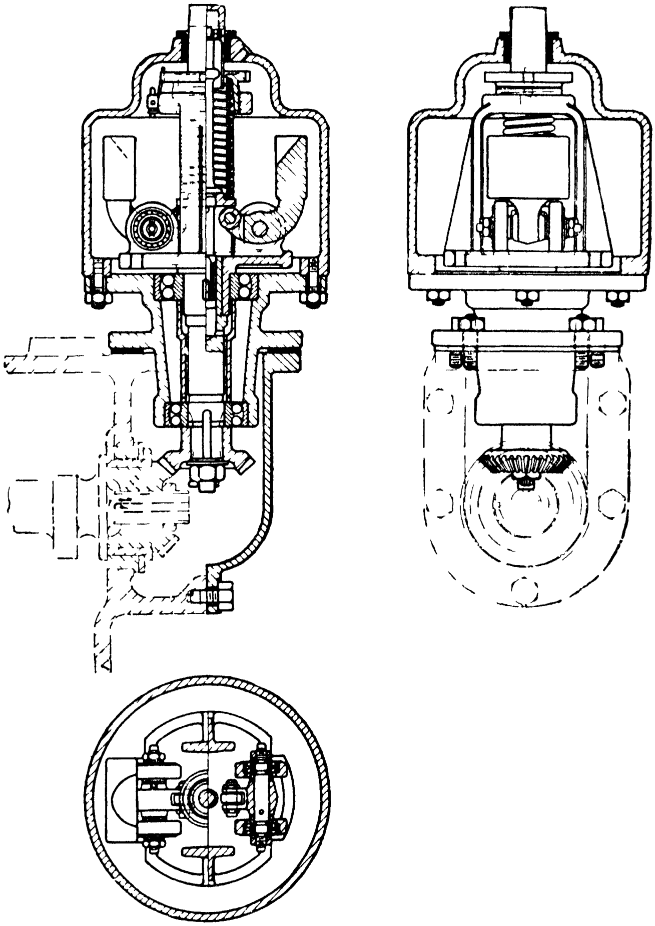


Fig 210.—Vertical Governor Arrangement

(2) The above test to be followed by a run of ten minutes at 1600 R.P.M. and not less than 150 B.H.P.

(3) Governor tests.

(4) A low-speed torque test, when each engine was required to develop not less than 55 B.H.P. at 400 R.P.M.

(5) The first engine by each maker and thereafter one in every fifty, as selected by the Inspector, were submitted to the following additional tests : –

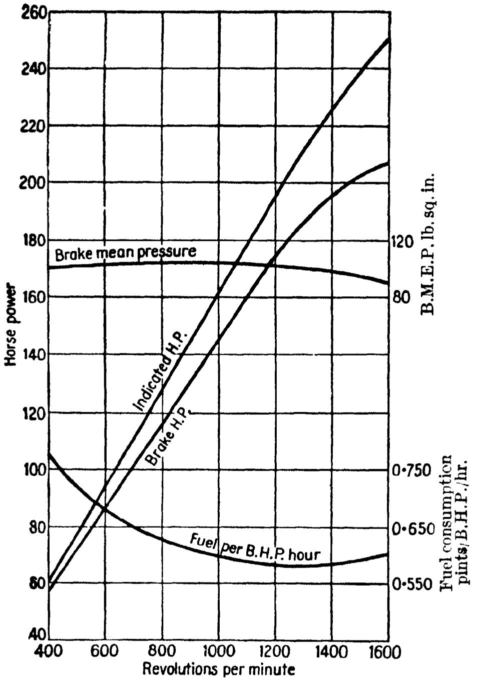


Fig. 211.—Performance Curves, 150 H.P. Engine

(*a*) A continuous full-load run of fifty hours, during which the conditions as to power, fuel and oil consumption were identical with those of the ordinary two-hour full-load test (1).

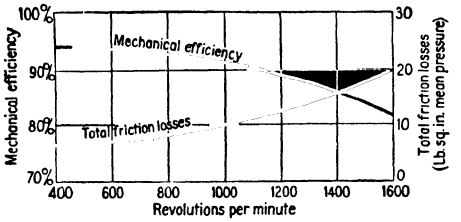


Fig. 212.—Mechanical Efficiency and Friction Losses, 150 H.P. Engine

(*b*) A tilting test, the engine to be mounted on a tilting table and tilted through an angle of 35° first in one direction and then in the other. When tilted at this angle the engine was required to be run for ten minutes at about 400 R.P.M. and no load; after this period the throttle was to be thrown wide open, when the engine was expected to open up firing regularly on all six cylinders without showing any smoke and without any oil leaking out of the base chamber.

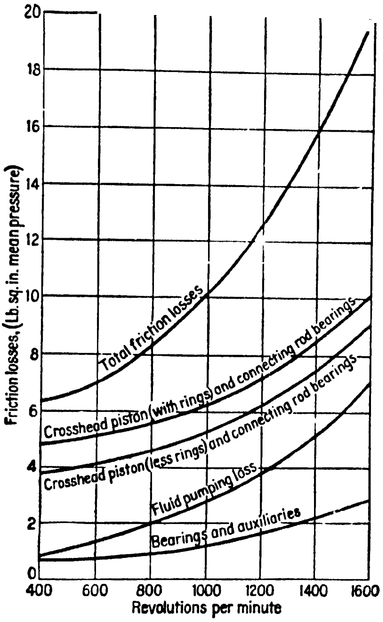


Fig. 213.—Mechanical Loss Curves, 150 H.P. Engine

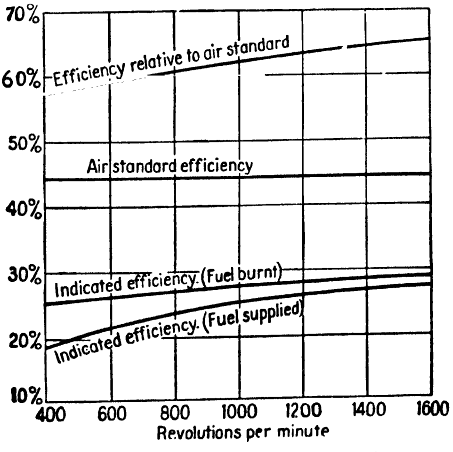


Fig. 214.—Efficiency Curves, 150 H.P. Engine

The useful life of a fighting tank was at first so short that an endurance between overhauls of 100 hours was considered ample, but, as might be expected, the engines in actual practice were called upon for an endurance of very much more than the 100 hours originally specified, and at least four instances were reported of engines having run 1400 hours at full speed without requiring or receiving any overhaul beyond the ordinary routine adjustments. Moreover, being of a convenient size and speed, they were used very largely for driving electric generators for supplying light and power to large camps, field workshops, &c., in which service their hours of running were naturally very much longer.

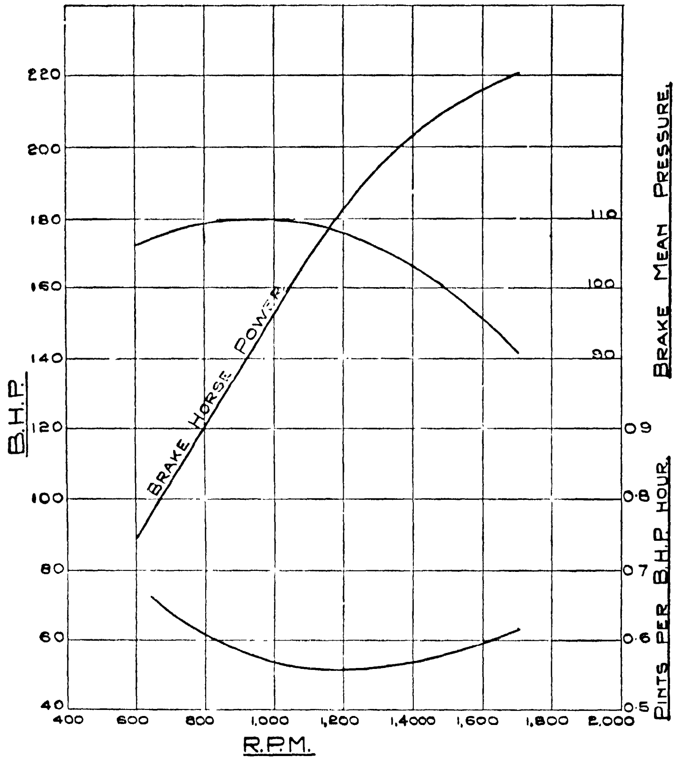


Fig. 215.—Power and Consumption, 150 H.P. Engine

The preceding curves (figs. 211–214) give in full the average performance of these engines. Fig. 211 shows the indicated and brake horse-power, also the brake mean pressure developed at speeds ranging from 400 R.P.M. up to 1600 R.P.M. The brake horse-power and torque curves are the mean of a large number of tests carried out by the different engine-makers, and may be taken as fair average results. Fig. 215 shows the results obtained from a particularly good example after the conclusion of its 50-hour full-power test. Fig. 213 shows the mechanical losses, which were determined in detail and with considerable accuracy by means of a swinging field dynamometer. All engines on completion of their official full-power run were motored for a short period to determine their mechanical efficiency, and the total mechanical losses were found to agree very closely with the sum of the several detail losses shown in the above curve. Further, in a few instances, tests for mechanical efficiency were carried out by the method employed by Morse, of cutting out one cylinder at a time while the engine is running on full load. These tests also showed very close agreement. All the test sheets showed that the mechanical efficiency, as arrived at by the motoring test, was remarkably uniform

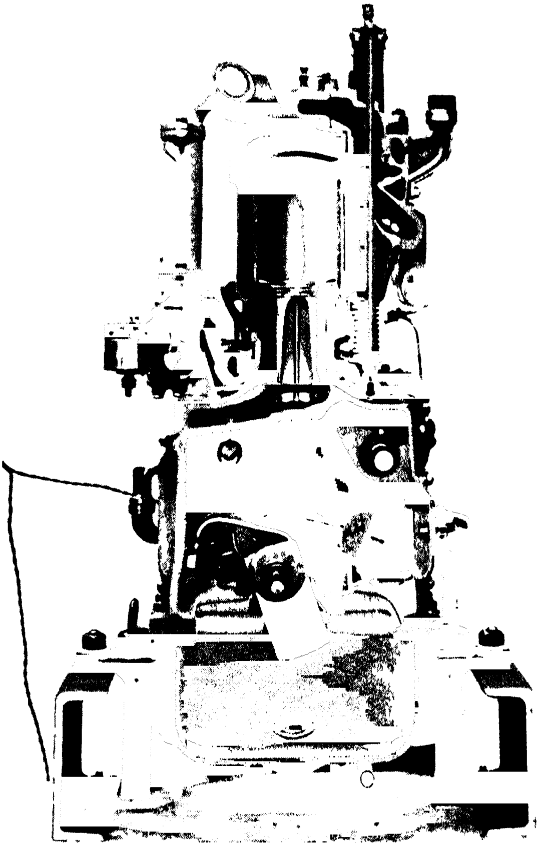


Fig. 216.—Sectional Model of 150 H.P. Engine

over a wide range of engines, a variation of 1 per cent in the mechanical efficiency figure being very exceptional.

Fig. 214 shows the thermal efficiency and the efficiency relative to the air standard; two efficiency curves are shown—(1) based on the fuel burnt, and (2) based on the fuel supplied. The efficiency based on the fuel supplied is calculated directly from the known fuel consumption. The efficiency based on the fuel burnt is arrived at by calculating back from the mean pressure actually obtained in the cylinder, and the difference between these two curves represents the loss due to imperfect carburation and distribution.

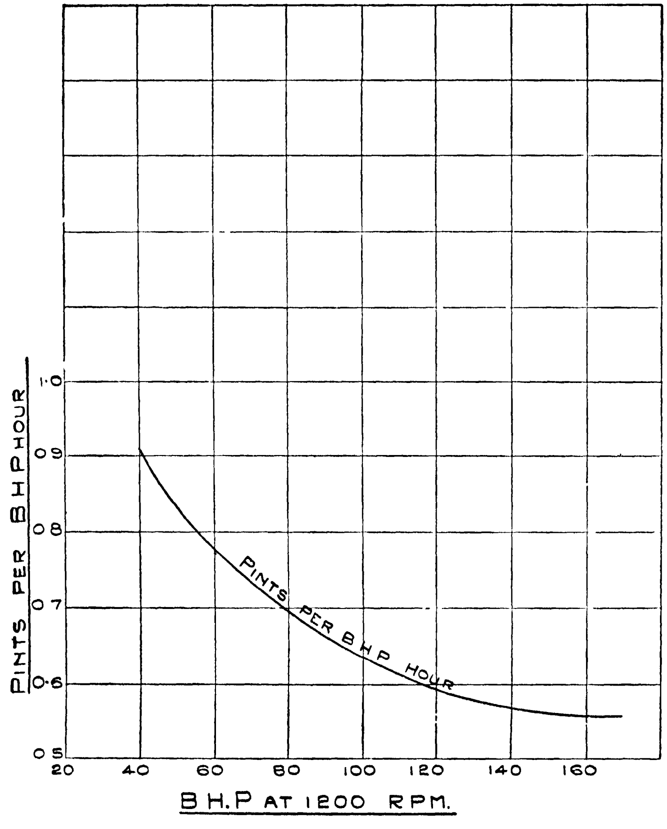
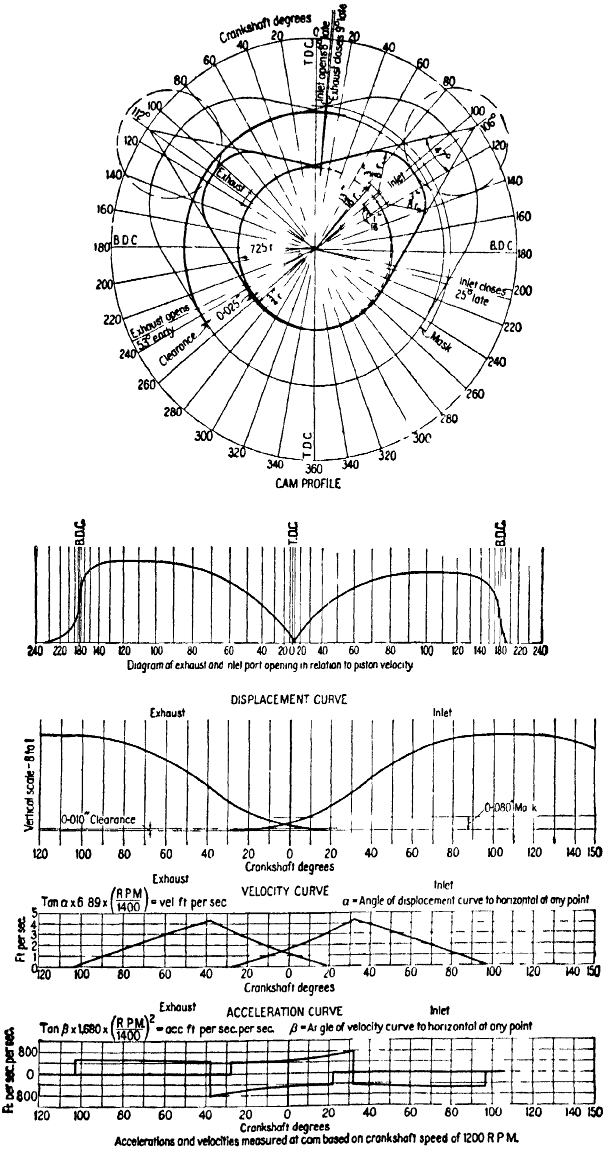


Fig. 217.—Throttle Curve, 150 H.P. Engine

Fig. 215 shows the power and consumption at full throttle and varying engine speeds.

Fig. 217 shows the fuel consumption at varying loads when running on the governor at speeds ranging between 1200 and 1300 R.P.M.

Fig. 218 shows in detail the cam formation and valve timing.



Accelerations and velocities measured at cam based on crankshaft speed of 1200 R P M.

Fig 218. —Cam Details

The following table taken from a sample test-sheet gives the heat distribution :—

Calibration Test, 150 H.P. Tank Engine

Duration of test.—Ten hours.
Fuel.—Shell spirit (specific gravity 0·725).
Lower heating value of fuel, 18,600 B.Th.U.s per lb.
Air standard efficiency—44·4 %.
Mechanical efficiency.—87 %.

Mean Results of Last Eight Hours of Test.

Brake horse-power	162·9
Fuel (lb./b.h.p.-hour)	0·554
Brake thermal efficiency	24·7 %
Indicated horse-power	187·0
Indicated thermal efficiency	28·4 %
Relative efficiency (per cent of air standard)	64·0 %
Heat loss to jackets (B.Th.U.s per hour)	418,000
Heat to indicated work	28·4 %
Heat to cooling water	24·9 %
Heat to exhaust, radiation, etc.	46·7 %

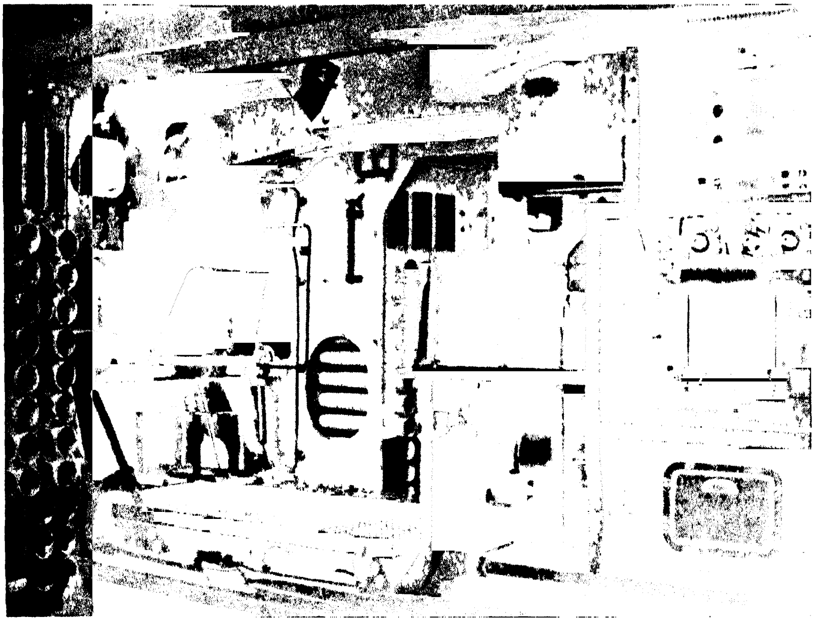


Fig. 219.—Interior of Mark V Two-star Tank

Fig. 219 shows the installation of the engine in the Mark V two-star tank, while fig. 220 gives an exterior view. (In this model the larger 225 H.P. engine was used.)

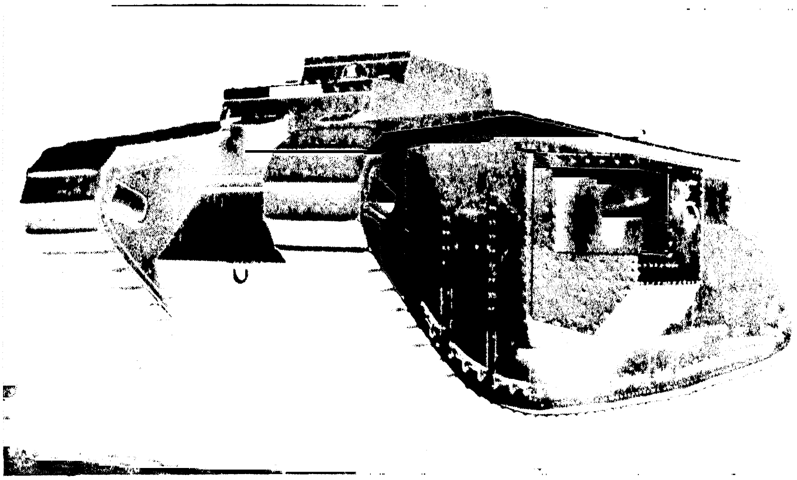


Fig. 220.—No. 106, Mark V Two-star Tank

The leading dimensions of these engines and general data are given in the following tables :—

Number and arrangement of cylinders ..	Six, vertical, separate.
Bore	5·625 in.
Stroke	7·500 in.
Stroke/bore ratio......................	1·323 : 1.
Area of one piston	21·85 sq. in.
Total piston area of engine	149·10 sq. in.
Swept volume of one cylinder	186·40 cu. in.
Total swept volume of engine	1118·10 cu. in.
Volume of clearance space	55·90 cu. in.
Compression ratio	4·34 : 1.
Normal b.h.p. and speed	165 b.h.p. at 1200 r.p.m.
Piston speed	1500 ft./min. (25·0 ft./sec.).
Indicated mean pressure	110·6 lb. sq. in. (187·5 i.h.p.).
Mechanical efficiency	88 %.
Brake mean pressure	97·3 lb. sq. in.
Fuel consumption	0·636 pint/b.h.p.hr. (s.g. 0·730). 0·580 lb./b.h.p.hr. (18,600 B.T.U./lb.)
Brake thermal efficiency	23·6 %.
Indicated thermal efficiency	26·8 %.
Air standard efficiency	44·40 %.
Relative efficiency	62·0 % (fuel burnt). 60·4 % (fuel supplied).

Gas Velocity, Valve Areas, etc.

Gas Velocity (ft. per sec.)—

Choke tube	353·0.

Carburettor body	168·8.
Vertical induction pipe	165·2.
Induction manifold	156·3.
Inlet port	109·6.
Inlet valve	168·3 (*e*). 130·8 (*t*.),
Exhaust valve	148·25.
Exhaust port	140·3
Exhaust branch pipes	140·3.
Exhaust manifold	126·6.
Cross Sectional Area (*sq. in.*)—	
Choke tube	1·760.
Carburettor body	3·680.
Vertical induction pipe	3·760.
Induction manifold	3·976.
Inlet port	5·672.
Inlet valve	3·690 (*e*). 4·750 (*t*).
Exhaust valve	4·190.
Exhaust port	4·430.
Exhaust branch pipes	4·430.
Exhaust manifold	4·910.
Diameter (*in.*)—	
Choke tube	1·496 in. (38 mm.).
Carburettor body	2·165 in. (55 mm.).
Vertical induction pipe	2·1875 in.
Induction manifold	2·250 in.
Inlet port	2·6875 in.
Exhaust port	2·375 in.
Exhaust branch pipes	2·375 in.
Exhaust manifold	2·500 in.
Weight of piston, complete with rings and gudgeon-pin, etc.	7·25 lb.
Weight per sq. in. piston area	0·292 lb.
Weight of connecting-rod, complete with bearings, etc.	10·70 lb.
Total reciprocating weight per cylinder	10·82 lb.
Weight per sq. in. piston area	0·435 lb.
Length of connecting-rod	16·00 in.
Ratio connecting-rod/crank throw	4·27 : 1.
Inertia pressure, top centre	82·2 lb./sq. in. piston area.
Inertia pressure, bottom centre	51·0 lb./sq. in. piston area
Inertia pressure, mean	33·3 lb./sq. in. piston area.
Weight of rotating mass of connecting-rod	7·13 lb.
Total centrifugal pressure	1094 lb.
Centrifugal pressure, lb. per sq. in. piston area	44·4 lb./sq. in.
Mean average fluid pressure, including compression	43·0 lb./sq. in.
Total loading from all sources, lb./sq. in. piston area	109·5 lb./sq. in.

Diameter of crank-pin	2·875 in.
Rubbing velocity	15·04 ft./sec.
Width of big-end bearing	2·25 in. (effective).
Projected area of big-end bearing	6·47 sq. in. (effective).
Ratio piston area/projected area of big-end bearing	3·84 : 1.
Mean average pressure on big-end bearing	121 lb./sq. in.
Load factor on big-end bearing	6330 lb. ft./sec.
Inlet Valve (*one per cylinder*)—	
Outside diameter	2·875 in.
Port diameter	2·6875 in.
Width of seating	0·0937 in.
Angle of seating	45°.
Radius under valve head	1·9375 in.
Diameter of valve stem	0·4995 in. in guide; 0·4985 in below guide.
Lift of valve (total)	0·5625 in.
Life of valve (effective)	0·4375 in.
Length of valve guide	4·375 in.
Clearance, valve stem in guide	0·0005 in.
Over-all length of valve	8·375 in.
No. of springs per valve	Two.
Free length of spring	(*a*) 5·875 in. (*b*) 5·8125 in.
Length of spring in position, no lift	(*a*) 4·000 in. (*b*) 3·8125 in.
Mean diameter of coils	(*a*) 1·703 in. (*b*) 1·120 in.
Gauge of wire	(*a*) No. 6 B.W.G., 0·203 in. diam. (*b*) No. 11 B.W.G., 0·120 in. diam.
Valve tappet clearance, cold	0·002 in.
Weight of valve, complete with spring, etc.	1·6234 lb. Valve bare, 0·8120 lb. Two springs, 0·6718 lb. Spring cap, etc., 0·1406 lb.
Weight of spring, bare	0·6718 lb. per pair.
Inlet valve opens (deg. on crank)	Valve leaves seat 29° early. Valve opens 8° late.
Inlet valve closes (deg. on crank)	Valve closes 25° late. Valve back on seat 59° late.
Material for valve	3% nickel steel stamping, case-hardened.
Exhaust Valve (*one per cylinder*)—	
Outside diameter	2·6875 in.
Port diameter	2·375 in.
Width of seating	0·156 in.
Angle of seating	45°.
Radius under valve head	0·875 in.
Diameter of valve stem	0·5595 in. in guide; 0·5475 in. below guide.
Lift of valve	0·5625 in.
Length of valve guide	7·125 in. (effective).
Clearance, valve stem in guide	0·003 in.

Over-all length of valve	12·8125 in.
No. of springs per valve	Two.
Free length of spring	(*a*) 5·875 in. (*b*) 5·8125 in.
Length of spring in position, no lift	(*a*) 4·000 in. (*b*) 3·8125 in.
Mean diameter of coils	(*a*) 1·703 in. (*b*) 1·120 in.
Gauge of wire	(*a*) No. 6 B.W.G., 0·203 in. diam. (*b*) No. 11 B.W.G., 0·120 in. diam.
Valve tappet clearance, cold	0·010 in.
Weight of valve, complete with spring, etc.	2·0934 lb. Valve bare, 1·281 lb. Two springs, 0·6718 lb. Spring cap, etc., 0·1406 lb.
Weight of spring, bare	0·6718 per pair.
Exhaust valve opens, degrees on crank	53° early.
Exhaust valve closes, degrees on crank	9° late.
Material for valve	3% nickel steel stamping, case-hardened.

Connecting-rod—

Length between centres	16·00 in.
Ratio connecting-rod/crank throw	4·27 : 1.
Little-end bearing, type	Chilled phosphor-bronze bush fixed in rod.
Little-end bearing, diameter	1·250 in.
Little-end bearing, length	1·625 in.
Little-end bearing, projected area	2·032 sq. in.
Big-end bearing, type	Bronze shell, lined white metal.
Big-end bearing, diameter	2·875 in.
Big-end bearing, length	2·6875 in.
Big-end bearing, projected area	6·47 sq. in. (effective).
Ratio piston area/projected area of big-end bearing	3·84 : 1.
Number of big-end bolts	Four.
Full diameter	0·500 in.
Diameter at bottom of threads	0·4375 in.
Total cross sectional area at bottom of threads	0·602 sq. in.
Total load on bolts, at 1200 r.p.m.	2659 lb.
Total load on bolts, at 1600 r.p.m.	4795 lb.
Stress per sq. in., at 1200 r.p.m.	4490 lb./sq. in.
Stress per sq. in., at 1600 r.p.m.	7965 lb./sq. in.

Crankshaft—

Length of complete shaft	66·00 in.
Cylinder centres	7·250 in.
Cylinder centres (centre pair)	8·000 in.
Outside diameter of crank-pin	2·875 in.
Inside diameter of crank-pin	1·4375 in.
Length of crank-pin	2·750 in.
Outside diameter of journals	2·875 in.
Inside diameter of journals	Solid.
Number of journal bearings	Seven.

Length, flywheel-end bearing	4·000 in.
Length, forward-end bearing	2·875 in.
Length, centre bearing.............	2·875 in.
Length, intermediate bearing	2·125 in.
Width of crank webs	3·500 in.
Thickness of crank webs	1·1875 in.
Radius at ends of journal and crank-pins	0·250 in.
Diameter of drilled oil-ways in shaft	0·250 in.
Weight of complete shaft	182 lb., includes thrust race pinion, starting clutch, nuts, etc. (no balance weights).
Material	Mild-steel forging.

It will be interesting and perhaps helpful to review the defects which revealed themselves in these engines. With over 4000 engines in service it is possible to discriminate between mere accidents and epidemic troubles.

Combustion Chamber.—The design of the combustion chamber, though excellent from the point of view of turbulence, and therefore of power output and efficiency, proved, as in the light of present knowledge might have been expected, rather bad from the point of view of detonation, despite the comparatively low compression ratio. Originally it was intended to operate the tanks only on aero-spirit. but later they were required to use the lowest grade of war spirit—an American fuel consisting almost entirely of the heavier fractions of the paraffin series. With this fuel detonation became severe when the engines were pulled down to a low speed with wide-open throttle.

Pistons.—The first few engines were fitted with sand-cast pistons in 88 per cent aluminium and 12 per cent copper alloy. These castings sometimes gave trouble owing to porosity of the metal at the point where the hollow trunk joins the head of the piston. Many pistons were rejected on this score, but a number in which the porosity did not appear on the surface and therefore was not detected by inspection were fitted to engines, and some of these broke away at this point, but since the connecting-rod was not released by such failures little or no further damage resulted therefrom. This defect was remedied completely by employing pistons cast in metal dies in all subsequent engines.

Cross-head Guides.—These were at first made of bronze lined with white metal, and proved quite satisfactory. Owing to the scarcity, or alleged scarcity, of bronze, the use of this material was eventually forbidden by the authorities, and cast iron was therefore substituted. The cast-iron guides gave a good deal of trouble owing to distortion after machining, and to meet this, since there was no time for "ageing" or annealing, it became necessary to allow rather a large working clearance, which gave rise to noise. Later the same copper-aluminium alloy used for the pistons was employed also for the guides, and this proved extremely satisfactory—quite equal to the white-metal-lined guides as regards wear, and better in so far as, since the clearance increased with temperature, it was safe to work with a very close fit.

Exhaust Pipes.—The exhaust manifolds and pipes radiated so much heat that it was found necessary, for the comfort of the tank crew, to jacket them with an air jacket through which a circulation of cold air was induced by means of a fan driven from the engine.

Crankshafts.—Owing in part to the small bearing area necessitated by the limits on the over-all length of the engine, and in part to the use of dead soft steel, the wear in the journals, and in particular the centre journal, was rather severe. So far as the actual fighting tanks were concerned, this was of little consequence, because the crankshaft easily outlasted the rest of the machine; but in the case of the tanks used for training, which ran all day and every day, and of electric-generating sets, this wear became troublesome, for it was generally necessary to regrind the crankshaft and fit new bearings after from 500 to 700 hours' running at full speed and 80 per cent load factor. It was therefore decided to fit balance weights on all engines destined for other than fighting tanks, a procedure which reduced the rate of wear to less than one-third, but which, by lowering the natural periodic speed of the shaft and by increasing the intensity of the oscillations, threw a heavy duty on the torsional vibration damper.

Apart from wear, the soft mild-steel crankshafts were very reliable indeed, and, out of over 4000 engines, no single instance of broken crankshaft was reported.

Vibration Dampers.– These were at first identical with those fitted to the Daimler engines. So long as the crankshafts were unbalanced, the duty on these dampers was very light indeed, and they gave no trouble. But so soon as balance weights were fitted and the dampers were called upon to function, the thin plates used in them soon cut away through the castellations and the damper wore out completely in about 50 hours' running. As the result of this experience the dampers were re-designed by Messrs. Gardner & Sons; two thick cast-iron plates with good substantial castellated bosses being fitted in place of the rest of thin steel plates: these proved perfectly satisfactory, the wear, even after 1200 hours' running with balance weights, being reported as almost negligible.

Lubrication System.—As explained previously—two scavenging pumps exhausted the oil, one from either end of the necessarily very shallow bedplate, and delivered it to a filter tank placed some 3 feet above the level of the crankshaft. A third force pump drew oil from the filter tank and delivered it to all the crankshaft bearings. All three pumps were operated from a single crank at the forward end of the engine, and had the same stroke; but the scavenging pumps were of slightly larger bore, and so could exhaust more rapidly than the force pump could deliver. It was found, however, that under certain conditions when the forward part of the engine was tilted up at the extreme angle the suction pump drawing from the after end failed to keep pace, with the result that oil gradually accumulated in the after end of the crankcase till it eventually ran out of the flywheel bearing. This was found to be due to the fact that while the force pump had a positive head of about 3 feet in its favour the scavenge pump had, under extreme conditions, a suction head of about 3 feet, while the length of suction pipe of about 5 feet was subject to a pulsating flow. Under these conditions the volumetric efficiency of the force pump exceeded that of the scavenge pump by more than the difference in volumetric capacity. This difficulty was cured completely by the provision of an air vessel on the suction side of the scavenge pump, thus maintaining a uniform flow in the suction pipe.

Again, it was found that, when the engine stopped in a certain position, oil could leak back from the filter tank through the bearings into the crankcase, sufficient in time to flood the latter. This occurred only when the machine was left standing for several days in very hot weather. To obviate this defect a cock was fitted in the oil pipe leading to the force pump, and, in order to render it impossible to start the engine with the oil supply cut off, this cock was combined with the magneto earthing switch.

Apart from these two minor difficulties, which were easily overcome, the lubrication system worked well; and although the scavenge pumps were considerably above the highest oil level, and under normal conditions their suction inlets also were above, yet they never failed to pick up the oil at once so soon as the engine was tilted and one or other of the suction inlets was drowned.

Valve Spring Caps.—In order to economize machining operations the valve spring caps were pressed out from sheet steel, and with a view to reinforcing the conical portion and prevent any risk of its being drawn over the taper wedges a steel wire ring was embodied in the pressing, as shown in fig. 221. In practice these pressed spring caps used to fail after a period ranging from 200 to 600 hours, the failure invariably occurring at the junction between the conical portion and the flat retaining face, as shown. The failure was due in part to fatigue and in part to the fact that, in spite of all precautions, the pressing was generally sharply nicked at this point. Though no doubt with a little more care in manufacture satisfactory pressings could have been made, it was decided, in view of the urgency, to employ spring caps turned from steel bar, which proved quite satisfactory.

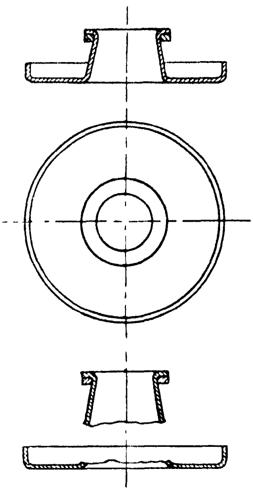


Fig. 221.

Valve Gear.—With a view to eliminating noise and wear in the valve gear the base circle of the cams was ground slightly eccentric in order to permit of a large working clearance at the time when it was essential that the valves should be closed, and gradually to take up this clearance preparatory to opening. This method is very satisfactory as a rule, but it is liable to abuse. In practice it was at first found very difficult to prevent the mechanics in charge from adjusting the clearance regardless of the position of the cam, with the result that the valves were lifted slightly from their seats long before the correct opening period. So far as the inlet valves were concerned no trouble arose, but a few cases of burnt-out exhaust valves were found to be due to this cause. The difficulty was eventually overcome by the issue of very full instructions for the adjustment of valve clearances and by rigorous inspection. Once the correct adjustment was fully understood, trouble with valves became almost unknown, and there is little doubt but that the immunity from trouble and the quiet running obtained were largely due to the use of cams with eccentric base circles. Apart from these troubles, of which all but the tendency to detonate and the heat radiated from the exhaust were quickly and easily overcome, these engines behaved very well. The system of hermetically sealing the crankcase and drawing all the air through the false top proved most effective in keeping the crankcase cool and the working parts clean.

The cross-head type pistons proved thoroughly effective both in eliminating the usual troubles with large aluminium pistons, in preventing smoke, and in yielding a very high mechanical efficiency.

With the steady growth in size and armament of tanks it became necessary to provide still larger engines, and another six-cylinder engine of the same general type but embodying all the experience gained with the 150 H.P. engine was next designed and produced. This engine is shown in the drawings and photos (figs. 222–226). Although about 800 of these engines were completed, yet, owing to delays in tank production, none of them saw any active fighting. Sufficient experience was, however, gained both on the test-bed and subsequently in tanks to prove that these larger engines were a very great improvement over the earlier type.

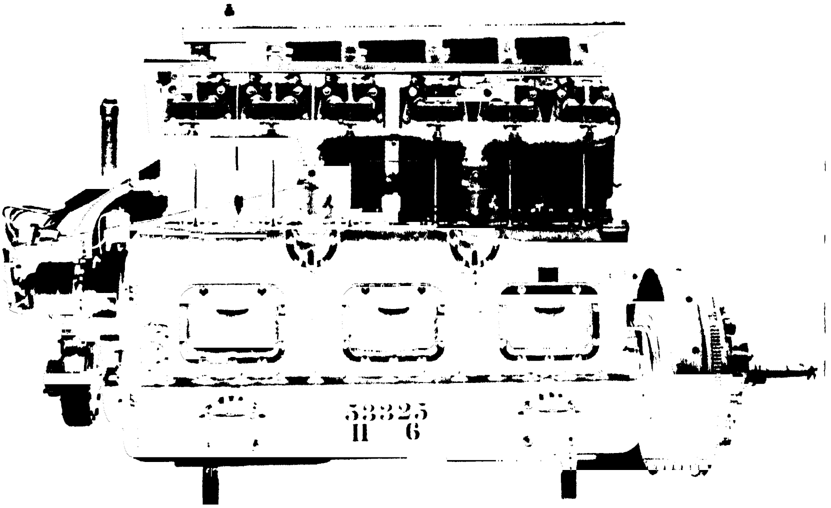


Fig. 222.—225 H.P. Engine, Carburettor Side

Rated normally at 225 B.H.P., they had a bore and stroke of 6¾″ and 7½″ respectively and developed 260 B.H.P. at 1200 R.P.M.

In order to obviate the defects of the 150 H.P. engines the following modifications were made:—

The combustion chamber was made in the form of a compact rectangular chamber with the sparking plugs placed as nearly as possible in the centre, as shown in fig. 226. This resulted in complete immunity from detonation under all conditions, so much so that the engines could be loaded down to two or three hundred R.P.M. with wide-open throttle without the least trace of detonation even on the lowest grade of American petrol, and this despite the fact that the cylinders were larger and the compression ratio as high as in the 150 H.P. engines.

The valves were duplicated and placed horizontally with the inlets on one side and the exhaust on the other. This arrangement of valves permitted of the exhaust ports being turned up vertically so that short straight exhaust pipes could be taken direct through the roof of the tank to a manifold fitted outside. Thus the length of exhaust pipe inside the tank was reduced to the very minimum, while all bends, &c., were avoided. The inlet valves were fitted in separate cages which could easily be removed without disturbing any of the pipe work, and the exhaust valves could then be

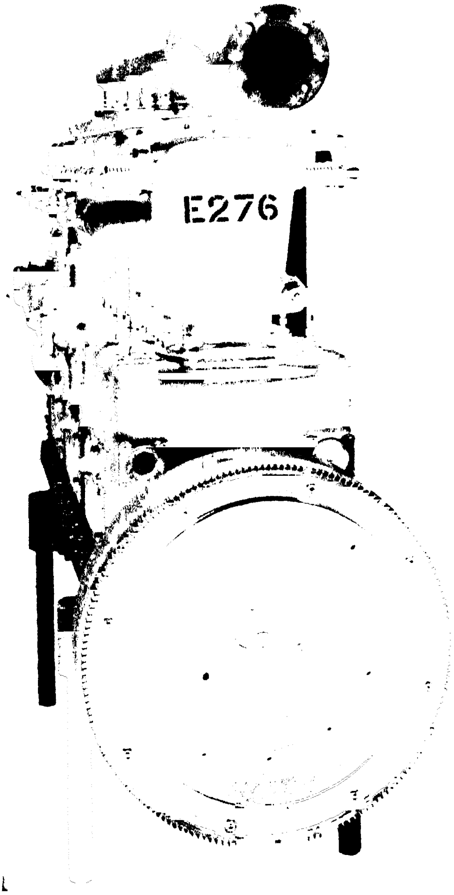


Fig. 223.—225 H.P. Engine, Flywheel End

Fig 224 —225 H.P. Tank Engines in course of erection at Messrs. Gardner's Works

withdrawn through the openings left by the removal of the inlet valves.

Die-cast pistons were used from the very start, and no single instance of piston failure was ever reported. There being no rigid restrictions as regards length, more liberal bearing surfaces could be and were provided, particularly as regards the crankshaft centre bearing. Balance weights and Messrs. Gardner's revised design of torsional vibration damper were fitted in all cases.

Fig. 225.—225 H.P. Engine, mounted for Tilting Test

The crankshafts were made from 0·35 carbon steel, and under these circumstances proved practically immune from wear. The

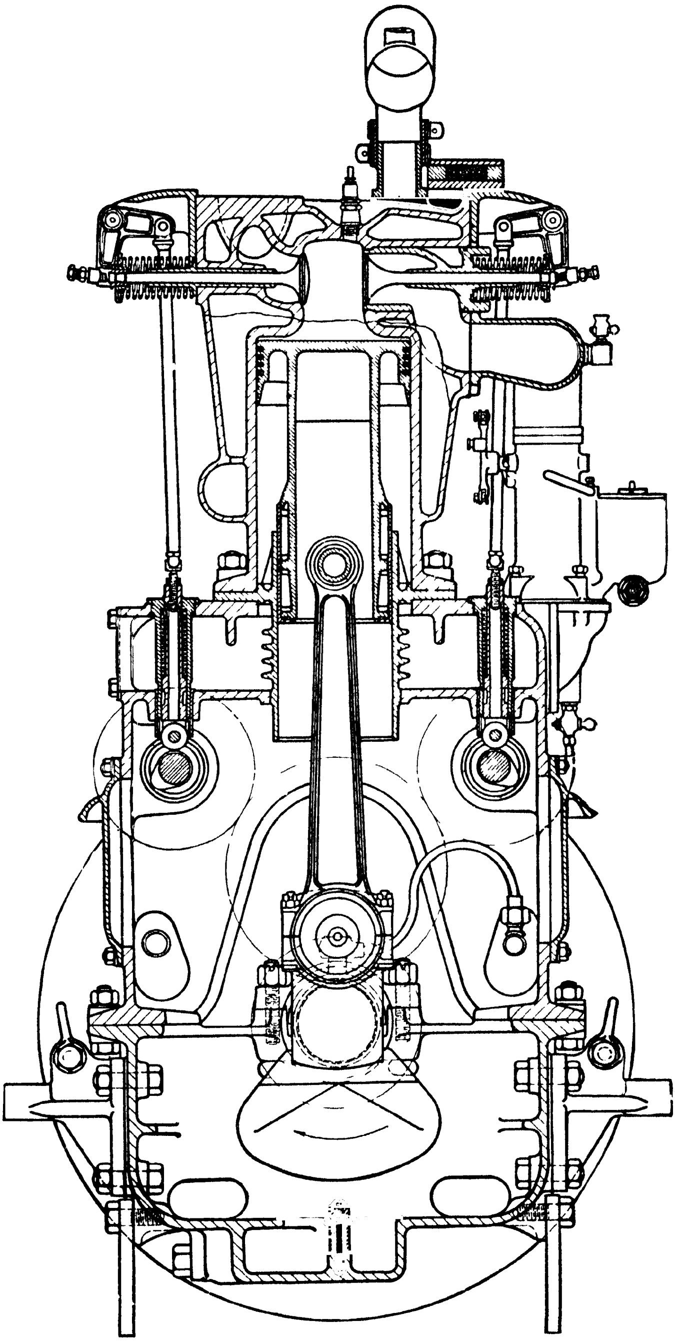


Fig. 226.—225 H.P. Tank Engine, Sectional Elevation

other defects which manifested themselves in the first batches of 150 H.P. engines were, of course, obviated by adopting the expedients explained above.

Although these larger engines never saw actual fighting service, yet they were submitted to very severe tests both in tanks and on the test-bed, the latter including full-power runs of 200 hours' duration. As may be supposed, with the experience of the smaller engines available before the design was commenced, these engines were an improvement on the 150 H.P. type in almost every respect. A very large number of them were later converted to run on gas, and were used in electric power stations in various parts of the country.

CHAPTER XII

HIGH-SPEED DIESEL ENGINES

In the years immediately following the war of 1914–18 attention was directed to the problem of so-called high-speed Diesel engines, that is to say, engines using relatively heavy and high boiling fuels, which cannot be vaporized outside the cylinder and have therefore to be injected into the cylinder in the liquid form, and which depend for ignition on the temperature of compression. The low fuel cost coupled with the very high thermal efficiency obtainable throughout the whole range of load renders such an engine very attractive in fields where fuel cost is a major item. In certain applications also, e.g. marine and commercial aircraft, the great reduction in fire risk resulting from the use of a fuel of high flash-point is a further important consideration in favour of the heavy-oil engine.

Ever since the Diesel engine first appeared towards the end of the last century, attempts had been made in various countries to produce a high-speed version, that is, a small engine operating at piston speeds in excess of, say, 1500 ft. per minute, but it was not until 1920 that any degree of success was attained.

It was not until the Royal Aircraft Establishment under the ægis of the Air Ministry tackled the problem that any really substantial advance towards the modern high-speed Diesel engine was made. To the Royal Aircraft Establishment is due the credit of having first shown the world, some twenty years ago, that a really high-speed heavy-oil engine could be produced which would not only run at a piston speed of well over 2000 ft. per minute, but that even at this high piston speed, a higher brake mean pressure and a higher over-all efficiency could be obtained than had ever been recorded previously by a slow-running Diesel engine of any size or type. It was these tests that dispelled the myth that Diesel combustion was hedged about by speed limitations and opened the way for the development work which, during the last eighteen years, has produced the modern high-speed Diesel engine.

Modern developments in high-speed Diesel engines have led to a wide departure from the slow-speed type and a much closer approach to the petrol engine, not only in mechanical design but also as regards the thermodynamic cycle on which they operate. That is to say, the idea of working on a constant-pressure cycle, never strictly adhered to even in the slow-speed engine, has in the high-speed engine gone entirely by the board, and instead a cycle is used which much more nearly resembles the constant-volume cycle of the petrol engine. The constant-pressure cycle or even any near approximation to it can be employed only in engines of relatively slow speed, and then only when the fuel is injected by means of blast air, since this method alone combines the requisite degrees of pulverization, of distribution and of turbulence. It is not, however, a desirable cycle for small high-speed engines, for its over-all efficiency is relatively low and its principal advantage, namely, a lower maximum pressure, is of less import when applied to a small piston. Moreover, even the air-injection engine may, owing to slight derangement, occasionally develop very high pressures and must, in consequence, be made robust enough to deal safely with them.

In any type of Diesel engine, whether of high or low speed, the major problem is that of bringing the fuel and air within the cylinder in contact with each other and, at the same time, of maintaining a high relative speed between the fuel particles and the air, in order to sweep away the products of combustion constantly as they are formed and to replace them with fresh air. The jet or jets of fuel must penetrate far enough through the dense atmosphere in the combustion chamber to reach the most outlying air, and at the same time the spray must be fine enough to spread out on all sides. This condition can be accomplished fairly readily when high-pressure blast air is used for injecting the fuel. By creating additional and violent turbulence the blast air assists in distributing the finely-pulverized fuel more or less uniformly throughout the combustion chamber. With so-called solid injection, i.e. without air blast, the problem is far more difficult and the conditions are conflicting. On the one hand, in order to reach the outlying air, we must have a fairly "hard" or penetrating jet; on the other hand, to give adequate dispersion of the particles of fuel, we require a "soft" or pulverizing jet. Thus solid injection starts with the disadvantage of having two apparently irreconcilable conditions to fulfil, and it is in the attempts to fulfil or circumvent these conditions that so much ingenuity has of late been displayed. No complete reconciliation has yet been found, but compromises have been attained which make the solid-injection engine at least as efficient as the air-injection engine and thus render possible the high-speed self-contained heavy-oil engine.

As stated previously, the thermodynamic cycle adopted in the high-speed Diesel engine as developed to-day resembles much more nearly that of the petrol engine than that of the large conventional Diesel engine. In reality, while there is a strong resemblance to petrol combustion in the early part of the process, during the final stages, when the rate of combustion depends on the rate of fuel entry, the process does in some degree approach the constant-pressure cycle.

Diesel and Petrol Combustion Systems.—Before proceeding further it is interesting to compare the two cycles:—

(1) In the petrol engine the whole of the fuel and air are intermixed very thoroughly, not only before ignition but even before admission to the cylinder. In the Diesel engine, air only is admitted to the cylinder during the suction stroke, and the fuel is injected at or just before the end of compression; hence the processes of intermixture and of chemical combination are co-existent.

(2) In the petrol engine ignition is brought about by the passage of a spark across the electrodes of the ignition plug; this leaves behind it a minute thread or nucleus of flame which spreads slowly at first, but with rapidly increasing velocity, to the envelope of mixture immediately surrounding it.

Under the conditions of indiscriminate turbulence which exist in any practical engine, the flame once fully established is torn and spread throughout the cylinder, with the result that the whole cylinder contents are inflamed with a rapidity many times in excess of that which would occur under stagnant conditions.

The Two-phase Character of Combustion in the Spark-ignition Engine.—The combustion process with spark ignition may be regarded as taking place in two distinct stages: (1) the building-up of a self-propagating nucleus of flame, and (2) the spread of the flame throughout the working fluid. The first phase occupies an appreciable interval of time, more or less independent of engine speed, during which there is no perceptible rise of pressure. The actual time occupied by the building-up of this flame nucleus depends on:—

(*a*) The chemical nature of the fuel.

(*b*) The mixture strength and hence the flame temperature itself.

(*c*) The temperature of the combustible mixture at the time of passage of the spark, and hence that of the surrounding envelope.

(*d*) The pressure of the combustible mixture at the time of ignition, for this controls the intimacy of contact between the inflamed and uninflamed particles.

The second phase, namely, the spread of flame throughout the main body of the combustible mixture, depends on:—

(*a*) The shape and form of the combustion chamber.

(*b*) The degree and nature of the turbulence whose function it is to spread the flame rapidly from zone to zone.

This second phase is a mechanical one pure and simple, and since turbulence increases pro rata with increase of speed, it follows that the time occupied by this stage varies with the speed of the engine and decreases as the engine speed is increased.

In a petrol engine, then, we may consider the combustion process as taking place in two distinct phases, the first of which may be termed the delay period, occupied by the building-up of a substantial nucleus of flame; during this period there is no perceptible rise of pressure. Under any given set of conditions as to fuel, mixture strength, compression pressure and temperature, this delay period occupies a substantially constant period of time irrespective of engine speed; therefore, as the speed increases the ignition timing must be advanced to compensate for the " delay ".

The second phase, that of rapid pressure rise, is determined by mechanical design and turbulence; under any given set of conditions its period is constant in terms of crank-angle but not of time. The author prefers to express this stage in terms of increase of pressure in pounds per square inch per degree of crank-angle. The two periods, of course, merge into one another, and, as explained in Chapter IV, it is the manner in which they merge that primarily determines the roughness or smoothness of running.

If the speed of a petrol engine is doubled, the delay period in terms of crank-angle will be doubled, or nearly so, but the period of pressure rise will remain the same. Thus fig. 227 shows typical diagrams on a time basis from a petrol engine running at 1000 R.P.M.' (above) and 2000 R.P.M. (below). In these diagrams the compression expansion line (when ignition is cut off) is shown dotted. It will be seen that in the lower diagram the angular movement from the passage of the spark until any appreciable rise of pressure occurs is approximately doubled, the values being of the order of 10° and 20° respectively, while the pressure rise occupies approximately 10° crankshaft degrees in each case, and amounts in this instance to 35 lb. per square inch per degree of crank-angle. The completion of the second phase marks the end of the combustion process so far as the petrol engine is concerned.

The Three Phases of the Diesel Combustion Process.—In the case of the Diesel engine with solid injection a stream of liquid fuel is injected into highly compressed and highly heated air. In this case the combustion process may be regarded as taking place in three distinct phases:—

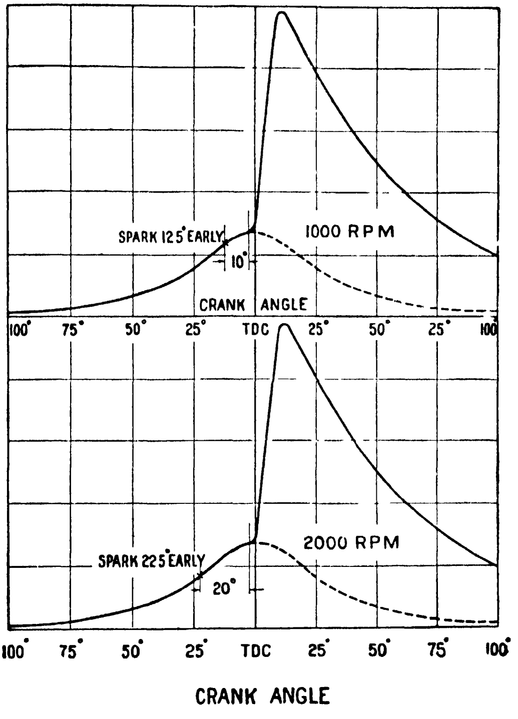


Fig. 227.—Diagrams showing Increase of Delay Period with Increase of Speed

(1) A delay period, during which fuel is admitted but either no ignition takes place or the ignition is restricted to some very localized nuclei. It seems probable that ignition first starts as soon as a few minute droplets of the injected fuel have had time to clothe themselves with vapour in the presence of sufficient oxygen to form a readily combustible mixture. While this is happening, more fuel is entering and a similar preparation for combustion by vaporization and mixing with oxygen is taking place. From the initial nuclei the flame spreads at first in much the same manner as in a spark-ignited petrol/air mixture during the first phase. As in the case of the petrol engine, the delay period appears to be constant in time for given

engine conditions and therefore independent of engine speed, and to depend on:—

(*a*) The nature of the fuel.

(*b*) The temperature and pressure of the highly compressed air.

(*c*) The fineness of pulverization, not so much of the fuel stream as a whole, but rather of the finest particles, for clearly an infinitesimal proportion of the total fuel charge will suffice to provide for these initial nuclei of flame.

Except for (*c*), the conditions as regards this first delay period are substantially the same for the Diesel engine as for the petrol engine, and in both cases the period of delay is constant in terms of time but not of crank-angle. The building-up of the nuclei is, however, much more rapid in the Diesel engine, owing to the higher pressure and temperature of the cylinder contents.

(2) The second phase, as in the petrol engine, consists in the mechanical spread of flame from the initial nuclei to the main body of the combustion chamber. Here again the conditions are similar to those of the petrol engine in that the rate of spread of flame and therefore the rate of pressure rise is dependent on turbulence and is constant in terms of crank-angle rather than of time. As in the case of the petrol engine, this phase can be speeded up to any desired extent by intensifying the turbulence. There is, however, the essential difference that, whereas in the petrol engine the whole of the fuel necessary to complete combustion is already present in the cylinder, in the Diesel engine only a proportion, and in some cases a small proportion, is present during phases 1 and 2. In these circumstances the pressure, though its rate of rise is of the same order as that of a petrol engine with similar conditions of turbulence, does not rise to anything like the high maximum which it would attain if sufficient fuel were present in the cylinder to combine with the whole of the oxygen. Clearly also phase 2 is influenced strongly by the amount of fuel present in the cylinder and the manner of its distribution throughout the air, for the mixture is no longer homogeneous as in the petrol engine, and the spread of flame during this period can take place only throughout those zones where a suitably proportioned fuel/air mixture exists at the time. The actual pressure attained at the completion of phase 2 depends on:—

(1) The extent of the delay period.

(2) The rate of injection of the fuel, which determines the amount of fuel available in the cylinder at the close of the delay period.

(3) The temperature and the pressure of the compressed air.

(4) The fineness of pulverization and the uniformity of distribution at this early stage in the proceedings.

Once the flame has spread through the combustion chamber the increase in pressure and temperature is so great, and hence the rate of combustion so accelerated, that the remainder of the fuel burns as it enters from the injector nozzle. Whether there is a further rise in pressure or a constant pressure is maintained depends upon the rate at which fuel is being delivered by the supply pump. During this, the third phase, the burning is under direct mechanical control and the pressure can be varied at will by varying the rate of injection, always assuming that the fuel is fed with an adequate supply of air throughout.

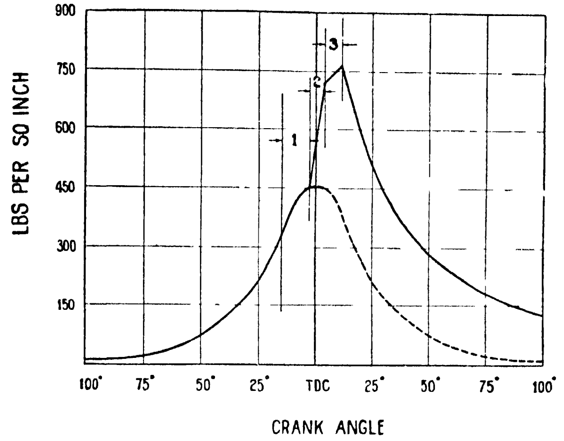


Fig. 228.—Diagram showing three phases of combustion process in Compression-ignition Engine

Fig. 228 shows a diagrammatic indicator diagram in which the three stages are shown as quite distinct, while fig. 229 shows an actual card taken from a small high-speed Diesel engine of 5½-in. bore by 7-in. stroke running at 1500 R.P.M., in which the three phases above referred to can be identified, though of course they are merged together to some extent.

Enough has been said by way of introduction to bring out the salient points of difference between the petrol and the Diesel processes of combustion. It may be useful now to examine some of these points a little more closely.

Influence of various Factors on the Combustion Process.—In the petrol engine the fuel is highly volatile and of high self-ignition temperature; combustion is initiated by touching off by means of a spark, and the temperature reached by that part of the charge awaiting inflammation is a critical feature in the process of combustion.

If after combustion has been started in one quarter self-ignition occurs elsewhere, such self-ignition gives rise to detonation. As has been shown, fuels of different chemical composition vary widely in their willingness to self-ignite. The chemical character of the fuel, then, is one of the determining factors in the process and decides the upper limit of compression ratio at which the engine can be safely operated. At the present time, with the fuels available in bulk, this ratio is of the order of 6·7 : 1 for the average-size cylinder.

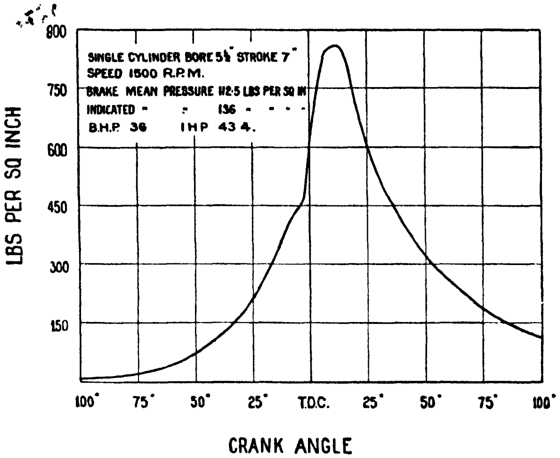


Fig. 229

In the Diesel engine, on the other hand, the fuel used is much less volatile and of lower self-ignition temperature. Combustion is initiated by the heat of compression without the aid of an igniter, and hence a compression ratio considerably higher than that of the petrol engine is used.

Compression Ratio.—In the nature of things as at present understood and practised there is a "no man's land" in the scale of compression ratio upon which it is not useful to encroach. Within the useful range at the lower end of the scale the incidence of detonation limits the compression ratio which can be employed in the petrol engine, even with the fuel least prone to detonate, to something like 9 : 1 for cylinders of average size, while the tailing off in the efficiency curve, the rise in maximum pressure and the increasing rapidity of combustion, and therefore the roughness of running, reduce the useful value to 7 : 1 or 7 : 5·1.

In the case of the compression-ignition engine, the minimum compression ratio which will enable the engine to start and at the same time enable combustion to be initiated rapidly and smoothly is round about 12 : 1.

Thus there is a wide neutral zone of compression ratio extending from about 7·5 to 12 : 1 which is not normally used for the internal-combustion engine of either the spark-ignited type or the compression-ignition type.

Within this zone in the case of the petrol engine the use of anti-detonating dopes is necessary, while in the case of the oil-engine, pro-detonating dopes, hot plates, &c., must be used.

As regards the upper limit for the Diesel engine, it does not pay to go beyond 20 : 1 even with small sizes of cylinders, partly because of the asymptotic character of the efficiency curve, and partly because the air lost in thin and inaccessible layers out of reach of the fuel becomes an increasingly greater proportion of the effective air volume. The maximum power output, which is mainly a function of the amount of air which can be burnt, therefore decreases as the compression ratio is raised. Moreover, the rise of pressure above compression must be reduced if the maximum pressure is to be kept within bounds; thus the final result is a loss in maximum power output without gain in efficiency.

From the above considerations it will be seen that for petrol operation the compression is for practical purposes limited to a ratio up to 7·0–7·5 : 1, depending on the fuel available and the design characteristics previously enumerated, while for compression-ignition operation the available range of compression ratio is from 12 to 20 : 1.

As a result of a wealth of experience and research data, certain significant facts about compression-ignition engines have come to light. In the first place, the combustion process as initiated between fuel and air by compression is much more rapid than is the case with petrol combustion. This is due to the higher charge temperature and density when combustion begins. The rapidity of burning, however, is much the same whatever fuel is used.

In the Diesel engine, so far as the actual combustion process is concerned, the nature of the fuel does not greatly matter. In this respect it is very different from the petrol engine, whose power output and efficiency are determined primarily by the octane number of the fuel. No such condition applies in the case of the compression-ignition engine. The higher the cetane value the shorter the delay period, and hence the smoother the running, but neither the power output nor the efficiency of the engine are affected appreciably by variations in cetane number, even over a very wide range. It is well that this should be realized in order to prevent a disproportionate regard being paid to small differences in cetane number for various fuels. The importance of 1 or 2 octane numbers in the case of petrol has created in some minds a similar attitude to cetane numbers.

Since the fundamental character of the process of combustion is much the same whatever the fuel, the problem of adapting this process to the C.I. engine is one of adjusting conditions to give optimum timing of events, and of controlling the rate and method of fuel input in such a manner as to keep the sequence of pressure changes within the limits imposed by practical considerations.

The problem therefore resolves itself into the mechanical one of using fuel and air to the best advantage by suitable combustion-chamber design, fuel spray and air movement. Broadly speaking, the relationships between these three factors are fairly well established, but there are a number of contingent variations in their application which complicate the problem. On this account success has been won only after exhaustive research into the effect of changes in such fuel-spray characteristics as penetration, atomization, and location and direction in relation to air movement; in the production of air movement, combustion-chamber shape and proportions, size and form of air throats, &c., naturally play an important part.

In the early Diesel engine attention was focussed on the fuel spray, and much research was expended in trying to divide up the fuel and spread it equally throughout the air charge. It can now be seen that severe limitations were thereby imposed both on the speed of the engine and on the power output, as controlled by the proportion of air burnt. The burden thus imposed on the fuel spray could be efficiently carried only at low speeds and at small fuel injections. Apparatus in these early days was not available for providing the high rate of injection and the greater degree of atomization and penetration required for high-speed operation, while the injection equipment available was incapable of delivering small fuel quantities at high rate and with uniform certainty, which is one of the essential requirements of the small high-speed oil-engine.

In these circumstances it is not too much to say that the successful use of fuel oil in the high-speed Diesel engine is due to two main lines of development:—

(1) The use of very rapid air movement to bring about fuel and air contact throughout the whole combustion process.

(2) The production of fuel-injection equipment capable of supplying and metering fuel in small quantities at high speed.

Just as the story of the high-speed petrol engine must be written largely in terms of air turbulence, the story of the high-speed Diesel engine turns on the use of organized air movement.

Just as it may be said that progress has depended on making use of air movement, so further progress will depend primarily on greater knowledge of the generation and pattern of air-flow in the combustion chamber and the relative effects of fuel and air-streams upon each other.

The Delay Period.—We are now in a position to discuss briefly some aspects of the various phases of the combustion process. Of the three phases referred to, (1) the delay period, (2) the period of uncontrolled burning, and (3) the final period depending on the rate of fuel injection, we have learnt how to exercise a large measure of control over the first, the delay period, while during the third phase we have almost complete control of the rate of burning and therefore of the pressure. It must be remembered, however, that anything done to alter the length of the delay period, while it does not alter the character of the second phase, may yet have a profound effect on its duration and hence on the extent of the uncontrolled pressure leap which precedes the third phase.

From the point of view of manipulating the pressure changes and of providing smoothness of running, we want to speed up the first two phases and carry out as much as possible of the combustion process in phase 3, that is, under the direct control of the rate of fuel injection.

It is not difficult to see that the longer the delay period the larger the amount of fuel which burns with uncontrolled violence under the conditions of phase 2, and the greater the Diesel knock. Moreover, during a long delay period scattered groups of fuel droplets may become dissociated from the main stream. These pockets of fuel in regions rich in oxygen may, during the uncontrolled combustion of phase 2, detonate in much the same way that detonation occurs in a petrol engine, though without the same ill-effects. In these circumstances the normal Diesel knock is overlaid with a high-pitched " pinging " noise.

The first thing to aim at, then, is a short delay period, which may be achieved in various ways:—

(1) By raising the compression ratio.

(2) By adding heat to the air during compression.

(3) By reducing the size of the fuel droplets and scattering them far and wide.

(4) By using a fuel of high cetane value.

For reasons already given, the raising of the compression ratio must not be pushed to extremes. Perhaps the best and most practical way of curtailing the delay is to fit heat-insulated members in the combustion chamber, which act as regenerators. By this means heat received during combustion is added to the air on compression. The method is somewhat limited in its effect, because all the heat taken in and stored by the member is obtained from the heat of combustion, and whereas on full load there is plenty of heat available, at light load or when running idle it is difficult to find enough. At full load, therefore, the delay period may become very brief, but after a period at light load it is liable to lengthen again. In spite of this limitation the insulated member is a most effective means of curtailing the delay, especially as the temperature of such members tends to rise appreciably with increase of speed, thus providing by suitable design an automatic control of the delay. A fixed injection timing may then be used throughout the speed range. Figs. 230 and 231 give curves showing the variation in temperature of such a hot member fitted in a combustion chamber of the type shown in fig. 255.

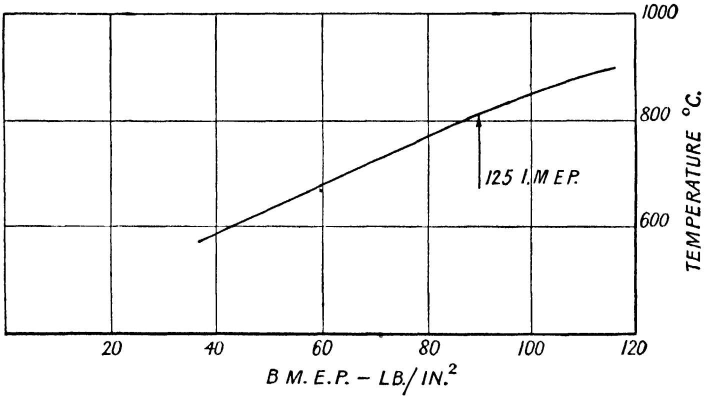


Fig. 230.—Variation in Temperature over the Load Range of a Hot Insulated Plug at 1500 R.P.M.

Fig. 232 shows how the delay period may be reduced as the speed increases, giving, for all practical purposes, a constant delay angle over the speed range. The variation of actual time of delay with speed is also given.

The delay period referred to is, of course, the time which elapses between the actual entry of the fuel as determined by timing the opening of the nozzle valve and the point at which the pressure in the cylinder rises above the normal compression line, this measured interval being assumed to correspond with the period between fuel entry and the initiation of combustion. The effect of adding heat to the air before it enters the engine is shown in fig. 234, in which it is seen that by raising the inlet air temperature from 19° to 84° C. the delay angle is reduced by 2 degrees. It must not be forgotten, however, that the volumetric efficiency is also dependent on air-inlet temperature.

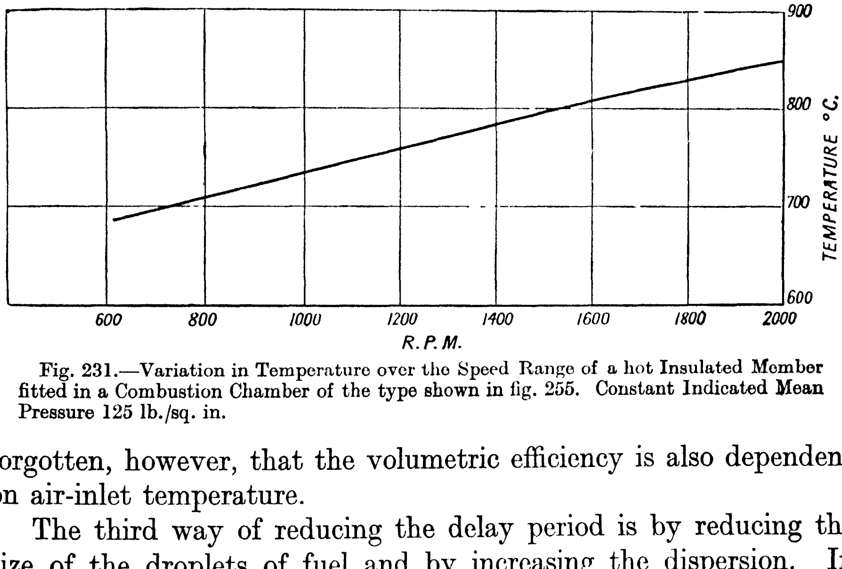


Fig. 231.—Variation in Temperature over the Speed Range of a hot Insulated Member fitted in a Combustion Chamber of the type shown in fig. 255. Constant Indicated Mean Pressure 125 lb./sq. in.

The third way of reducing the delay period is by reducing the size of the droplets of fuel and by increasing the dispersion. In practice, however, the amount that can be done in these directions is very limited because the required high degree of penetration prevents the use of a very fine spray, and a size of nozzle hole of the order of 0·010 in. represents the practical minimum limit for satisfactory operation for long periods of running.

The fourth method, that of using a fuel of high cetane value, is an effective way of reducing the delay period.

Measurement of the Delay Period and Rating of Diesel Fuels.—A great deal of valuable research work has been done by Boerlage, Broeze and others on methods of measuring ignition delay, with a view to setting up a standard of comparison for Diesel fuels. The rating scale in use at present is based, in much the same way as for petrols, on comparison with a mixture of cetane with a standard fuel of poor quality, the percentage of cetane providing the rating number of the fuel under test. The difficulties of accurate measurement of delay period from indicator diagrams are well known to those having experience of them.

Interesting data have been obtained by Dr. F. F. Butterworth on an E.6 variable-compression research engine installed in the author's laboratories. The method of the first series of tests given below was selected from a number tried, as a good practical way of distinguishing and measuring fuel quality in this engine.

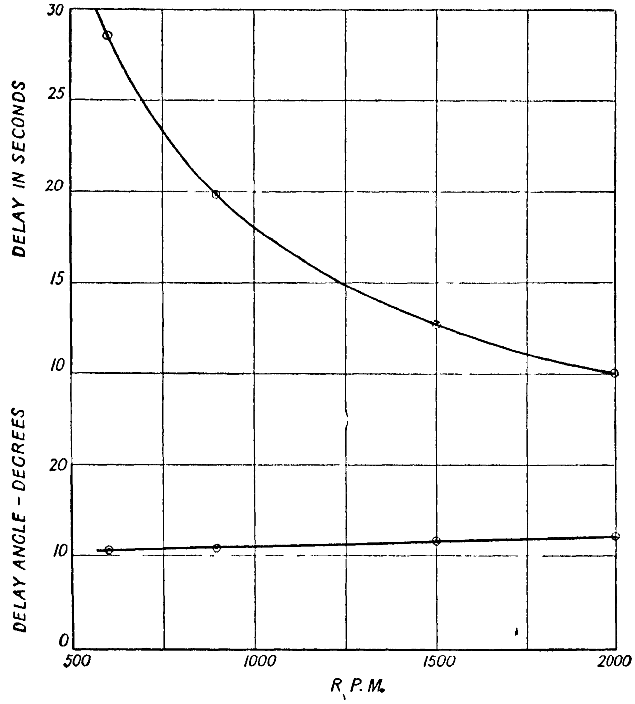


Fig. 232.—Curves showing Ignition Delay in relation to Speed, with insulated member forming lower half of Chamber. Whirlpool Type, C.R. 16. Fuel—40 cetane

The engine has been mentioned in an earlier chapter as a research unit for petrols. It is shown in figs. 13 and 14 with a cylinder head used for research on Diesel fuels. The available range of compression ratio of the engine thus equipped is 5–20. The two series of tests illustrated in figs. 233, 234 and 235 were carried out with the aid of a Sunbury Cathode Ray Indicator, and with engine conditions found by experience to give steady diagrams of nozzle valve lift and gas-pressure change. In the first series the compression ratio was kept fixed at 19·75 : 1 and the injection timing was varied over the widest possible range. The fuel quantity was set to give 90 lb. pér square inch I.M.E.P. at 1500 R.P.M., with the

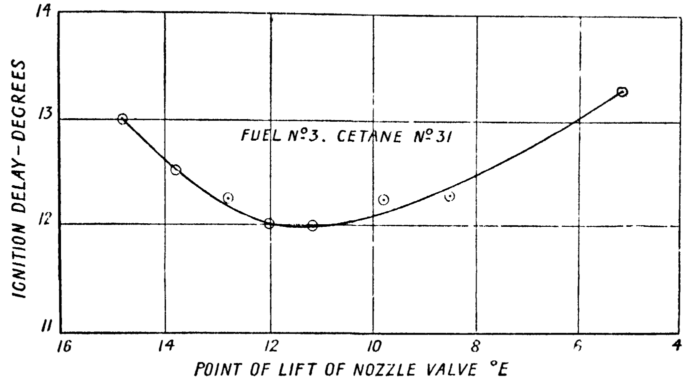


Fig. 233.—Variation of Ignition Delay with Injection Timing. 1500 R.P.M., C.R. 19·75. Fuel quantity to give 90 lb. per sq. in. I.M.E.P. with pressure rise at T.D.C.

rise of pressure from the compression line occurring at T.D.C. Fig. 233 shows the type of curve of ignition delay obtained in this way, while fig. 234 gives a summary curve drawn through the minimum

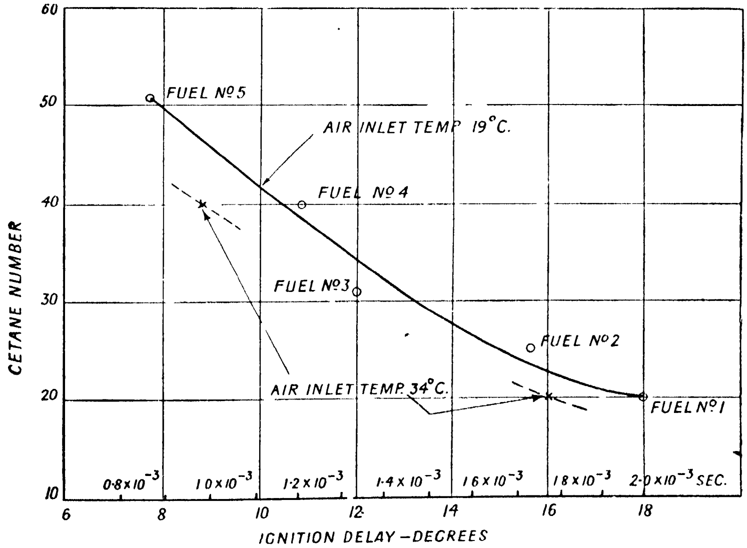


Fig. 234.—Relationship between Cetane Number and Ignition Delay Angle. Engine speed 1500 R.P.M. Compression ratio 19·75. Fuel quantity to give 90 lb. per sq. in. I.M.E.P. with pressure rise at T.D.C.

delay angles for each fuel. The conditions for minimum delay were determined for each fuel when ignition took place at T.D.C.

In the second series of tests the effect of compression ratio on delay angle was investigated. The method consisted in determining for a particular fuel the compression ratio for a given delay angle, the conditions being based on preliminary exploratory tests with the fuel of poorest quality. The compression ratio was then adjusted with each fuel tested to give the same delay characteristics. It is seen that the values all lie reasonably on a curve; thus fig. 235 gives the compression ratio for a standard delay angle plotted against cetane number.

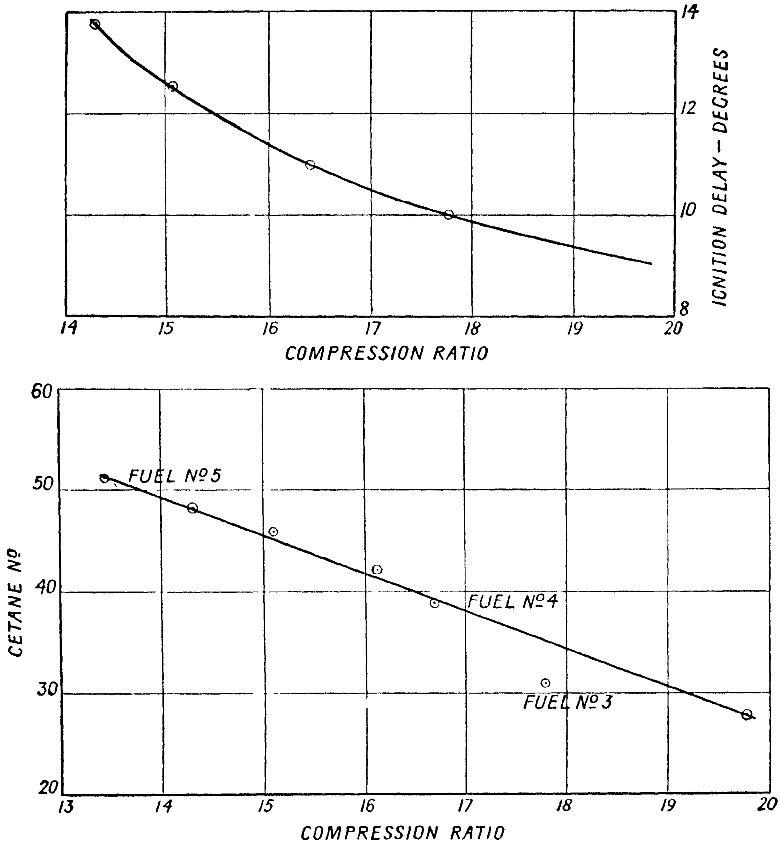


Fig. 235.—Variation in Ignition Delay with Compression Ratio

The range of fuel oils tested was as follows:—

Fuel 1	S.G. ·946	Cetane No. 20
„ 2	„ ·924	„ 25
„ 3	„ ·915	„ 31
„ 4	„ ·908	„ 40
„ 5	„ ·879	„ 51

The fuel oil in general use to-day in high-speed Diesel engines corresponds roughly with Fuel No. 5, but by taking precautions in design, fuels of the quality of No. 3, say cetane No. 30, may be used without detriment to the regular and smooth running of the engine, and usually with a slight gain in performance. Below this value, however, the running of the engine is uncertain and misfiring will set in with slight provocation. It has been the practice to specify a high-grade fuel for the modern high-speed Diesel, owing to the unpleasant effects in the way of nozzle troubles, rough running and poor combustion which have attended the use of fuel of poorer quality. This limitation hardly applies to-day, although, like the parallel rejection of high-gravity fuels in the early days of the petrol engine, such ideas and practices die hard.

A good deal has been said about reducing the delay period, but the reduction must not be overdone. In the extreme case in which there is no delay at all, a purely imaginary state of affairs, the burning would be localized and concentrated in the region of the nozzle, thus preventing a large proportion of the air from finding the fuel. The result would be very smooth combustion but a miserable power output. The best send-off is to provide a delay period which allows the fuel to penetrate to the far side of the chamber before ignition takes place.

As the delay period remains constant in time, unless controlled by some such artificial means as that previously mentioned, the higher the engine speed the greater the quantity of fuel admitted during this period, the rate of injection being proportional to crank-angle. Hence, also, the greater the quantity of fuel which burns during phase 2 and the higher the maximum pressure. From this it would appear, as indeed experience indicates, that under any given set of conditions as to rate of injection, temperature, &c., the higher the speed the higher the maximum pressure involved if the same efficiency is to be maintained. In other words, the higher the speed the greater the departure from the constant pressure and the nearer the approach to the constant volume cycle. Therefore, in order to keep down the maximum pressure, which would exceed practical limits at high speed, it is desirable either to inject the fuel, not at a constant rate, but rather gradually at first, so that not too much shall enter during the delay period, or to reduce the delay period, as the speed is raised, by the introduction of an insulated hot member.

It used to be argued that the combustion process in a C.I. engine, including the vaporization of the liquid particles, the subsequent mixture of this vapour with sufficient air for combustion, and finally the combustion of the air/vapour mixture thus formed, must necessarily take so long a period of time as to render the high-speed C.I. engine either entirely impracticable or at least so inefficient as to be of very little service. Such arguments have been shown to be entirely unfounded; not only have they been refuted by practical results, but they appear to be based on false premises. Except that a very minute quantity of vapour must be present to provide the initial nucleus of flame, there does not appear to be the slightest ground for supposing that the rest of the fuel must be vaporized completely before combustion can take place; on the contrary, it appears far more probable that combustion actually takes place from the surface of the liquid droplet which is fed with fresh oxygen and has the burnt products scoured away by the rapid passage of this droplet through the air. The really essential condition for high speed is sufficient relative motion between the drops of fuel and the air—a condition which can be met either by rapid movement of the fuel or of the air, or both (provided, of course, their direction is not the same).

The Second Phase.—The second phase of the process, which is the least accessible to adjustment of the three, may be left to take care of itself, provided there is proper control of the other phases.

With a shortened delay period it is found that the quantity of fuel awaiting combustion when ignition begins can be reduced sufficiently to make special provision of turbulence for its burning unnecessary, always provided a high degree of air swirl is in use. In these circumstances phase 3 becomes the next important consideration.

Phase 2, however, was important, before steps were taken to make it harmless, as the period responsible for Diesel knock, and in conditions of air stagnation in the chamber this may still be a nuisance. In any case, means must still be sought to provide a gentle rate of pressure rise during this phase when, under idling or light load conditions, the delay period is unduly prolonged.

The Third Phase.—The third phase, namely, that in which the fuel burns from the surface of the liquid droplets as they enter from the fuel injector, is the only one in which the designer has direct mechanical control over the rate of burning, and therefore over the pressure. It should be his aim to arrange things so that phase 3 shall be the main part of the combustion process, phases 1 and 2 being merely prologues. Phase 3 becomes possible only when the temperature and pressure have been raised by phase 2 to beyond a certain minimum. For phase 3 the all-essential condition is that there shall be sufficient relative motion between the droplets of liquid fuel and the air for each individual droplet to travel like a rocket, leaving behind it its trail of combustion products, proceed onwards to fresh fields of adventure, and thus to search out and combine with as large a proportion as possible of the oxygen in the combustion space. This condition can be met only by securing rapid relative motion between the fuel and the air. This rapid motion can be obtained in various ways:—

(1) By the use of very high injection pressures, thus giving to the fuel itself a high initial velocity. This method is, however, somewhat handicapped by the fact that the speed at which a liquid jet can issue from an orifice is limited; for beyond a certain point increase of pressure results merely in breaking up the droplets rather than in increasing their velocity; moreover, the initial velocity gained by high initial pressure is soon dissipated in the dense atmosphere. It is open also to the practical objection that very high pressures in conjunction with the somewhat elastic properties of the fuel oil and its pipe line add greatly to the problem of timing and metering. The general principle of this system is illustrated in fig. 236.

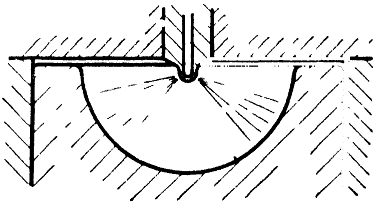


Fig. 236

(2) By the use of a so-called pre-combustion chamber, which consists of a small chamber separated from the main combustion space and communicating with it by a number of small holes. Fuel is injected into this small antechamber, where phases 1 and 2 develop; during phase 2 the rapid rise of pressure projects the remaining unburnt fuel, along with highly heated air and some products of combustion, into the main combustion chamber at a velocity far in excess of that which can be obtained by any purely mechanical injection of the liquid fuel. This method has certain obvious advantages; in contradistinction to (1), the engine speed is not limited by the speed of the injected liquid, nor is there the same difficulty in distributing the liquid particles throughout the air, since any number of holes can be provided in the " pepper castor " interposed between the pre-combustion chamber and the main chamber, whereas the number of jets for liquid fuel is strictly limited by mechanical con-

siderations. On the other hand, the loss of heat entailed by passing the products through a number of small holes at so high a temperature and velocity is very great and curtails the efficiency and power output very seriously. This principle is shown diagrammatically in fig. 237.

(3) By producing organized unidirectional flow of the air itself and then projecting the fuel across the air-stream. The highly heated air then sweeps past each drop of fuel as it enters, carrying away the products of combustion as they are formed, and continually bringing up fresh or only partially contaminated air to the attack. Since the velocity of the air may in any case be enormously greater than that of the liquid fuel, and is at all times proportional to the engine speed, further, since it can be maintained almost constant throughout the combustion process, this system is, in the authors' opinion, by far the most promising of the three. It has also very important practical advantages, which will be dealt with later. This principle is shown diagrammatically in fig. 238.

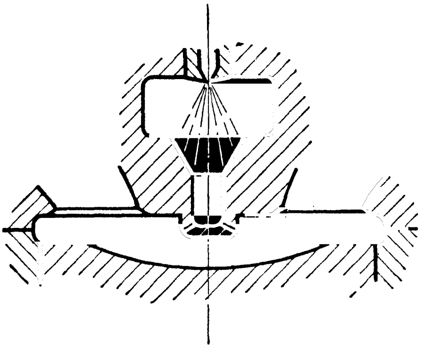


Fig. 237

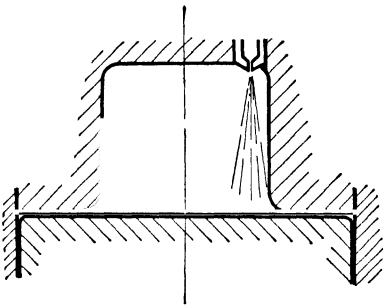


Fig. 238

The problem of phase 3 is very closely related to that of carburation in the average petrol engine; in both, the aim is to distribute the particles of liquid fuel uniformly throughout the air, the chief difference being that in one case they evaporate and in the other they burn from the surface; but this difference does not affect the mechanical problem, which remains much the same in either case.

The three systems referred to above correspond in effect with three methods of carburation, namely:—

(1) By projecting a spray of liquid petrol under pressure into the air—a method which has been tried from time to time, but not adopted on any considerable scale.

(2) By the use of a " diffuser " jet, as in the Claudel type of carburettor—this corresponds with the pre-combustion chamber system.

(3) By concentrating all the air at a high velocity past the fuel jet by means of a choke tube, as in most ordinary carburettors; this corresponds with the last system described, namely, that of air swirl or unidirectional flow.

Yet a fourth method, and one which does not fall strictly into any of the above categories, is that known as the " Acro " system. In this system practically the whole of the air for combustion is compressed into a chamber, usually in the piston and communicating with the rest of the cylinder through a narrow restricted neck. The fuel injector is so placed that the jet of liquid is aimed at the centre of this narrow neck, as shown in fig. 239. Fuel injection starts slightly before top dead centre and is continued relatively slowly. Phases 1 and 2 of the combustion process are enacted more or less inside the chamber, where the indiscriminate turbulence is very intense; but since the fuel injection is timed rather later and is spread over a fairly long period, very little fuel has actually been injected before the piston starts to descend, when the highly compressed air within the chamber commences to rush out through the neck. Phase 3 is therefore enacted outside the chamber under conditions of intense air-flow, for during this stage air is rushing helter-skelter out of the chamber and, in so doing, meets the injected fuel which is approaching to meet it. The combustion process during phase 3 therefore resembles that of the last system described, but with this essential difference, that in the " Acro " system phase 3 can only occur while the piston is descending fairly rapidly, that is, relatively very late; hence the expansion ratio is curtailed and both the power output and the efficiency are somewhat limited. In this system it is difficult to determine just where combustion takes place, but it probably oscillates rapidly about the open neck of the bulb; the bulb itself is quite uncooled, and since it does not attain a high temperature, it is quite clear that the bulk of the combustion process must take place outside it and beyond the neck; on the other hand, as the pressure rises, even though the piston is descending, there

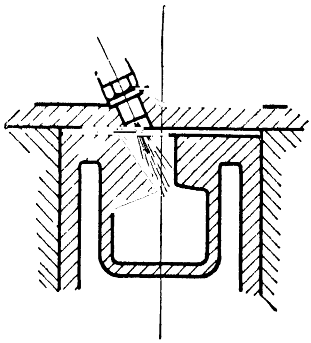


Fig. 239

must be reversals, probably rapidly alternating reversals, of flow between the bulb and the cylinder.

We shall now compare again the conditions in a petrol engine and a high-speed solid-injection Diesel engine. Phases 1 and 2 are common to both, except that in the petrol engine the whole process is consummated in phase 2, while the equivalent of phase 3 is antedated, and is enacted not in the cylinder but in the carburettor.

In the petrol engine we need indiscriminate turbulence in the combustion chamber and processional flow in the carburettor; in the Diesel engine we need both within the combustion chamber itself, and since the conditions are conflicting, we have to make the best compromise we can.

Mechanical Considerations.—The compression-ignition engine is rapidly replacing the petrol engine for commercial road-vehicle operation. At first it was natural that this should take place in the larger sizes, but development has now reached a point where the compression-ignition engine can be applied to any size of vehicle, and is in fact being thus employed in Continental countries where the tax on Diesel oil is not so high as that on petrol. To compete to-day in the fields of the lighter vehicles, speeds up to 4000 R.P.M. are required for the very small class using engines of 1·5–2 litres capacity and developing 40 to 50 H.P., and up to 2800–3000 R.P.M. in the somewhat heavier classes requiring 60 H.P. In the heavier vehicles for which upwards of 100 H.P. are needed, the engine must run up to 2400–2500 R.P.M., involving piston speeds of over 2000 ft./min.

A spate of mechanical problems confronts the designer who seeks to attain these high speeds. To-day the Diesel engine is expected to operate with smokeless exhaust at all loads over a range of speed of over 8 : 1 and with a fixed injection timing.

This wide range at once introduces the difficulty that the volumetric efficiency of the engine as an air-pump and that of the fuel-pump both vary with speed, but in different ways. The volumetric efficiency of the fuel-pump is much more nearly constant than that of the engine. To avoid exhaust smoke at all speeds there must always be an excess of air, and therefore in the ideal condition the air/fuel ratio should remain constant throughout at a value allowing a certain air quantity over and above that required for chemically correct mixture.

Fig. 240 shows two typical volumetric characteristics for the engine and for the fuel-pump, and clearly indicates that to provide the ideal condition the volumetric efficiency of the engine must remain virtually constant over the whole speed range. This cannot be, as at low speed the quantity of air per cycle is reduced, partly by reverse air-flow during the first part of the compression stroke, when the inlet valve must remain open for filling at higher speeds, and partly because the air charge picks up relatively more heat during entry to the cylinder at low speeds. Again, at high speeds the breathing capacity of the inlet valve falls, owing to wire-drawing.

Some sacrifice of the attainable maximum torque must therefore be made in the middle of the speed range, in order to prevent excessive smoke appearing in the exhaust at the low- and high-speed ends. Clearly the wider the speed range demanded the greater the sacrifice.

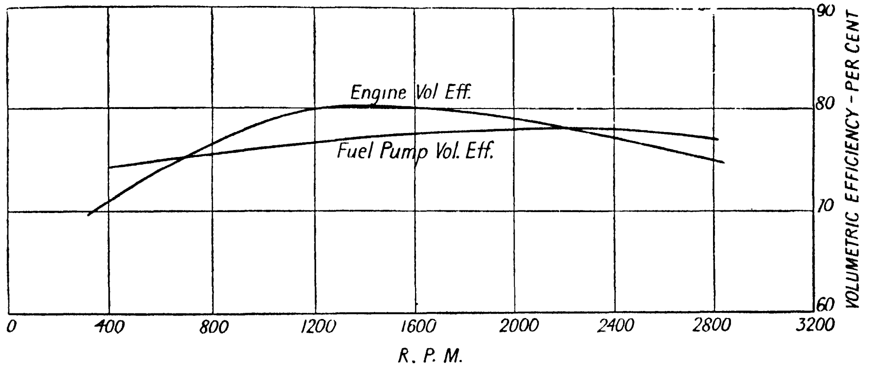


Fig. 240.—Variation of Engine and Fuel Pump Volumetric Efficiencies with Speed

The aim of the designer, then, is to match these two curves as nearly as possible, and while a little may be done in this direction by attention to the injector system, greater assistance is obtained by removing as far as possible the restriction of the inlet valve. A considerable degree of flattening of the volumetric efficiency curve may be achieved by making the inlet valve as large as space will allow and by closing it as early as possible to maintain low speed filling without penalizing the inflow at high speeds. Fortunately, with the high compression used in C.I. engines we can use a relatively small exhaust valve, thus giving room for a large inlet. In any case the greater part of the exhaust products passes out while the piston is in the region of bottom dead centre, so that normally there is but a very low back pressure, if any, and this, in any event, has little effect, thanks to the relatively small capacity of the clearance space.

Mention has been made of the use of air movement in the clearance space, and to-day nearly all Diesel engines depend on the practice of using either intense air turbulence or intense air swirl to bring about good combustion.

There are three ways in which such turbulence can be generated. A rotary movement may be imparted to the air:—

(1) While it enters the cylinder.

(2) During the compression stroke.

(3) During the actual combustion process.

Broadly speaking, air swirl created by method (1) yields the best thermal efficiency, but at a heavy cost in volumetric efficiency, except in two-cycle or sleeve-valve four-cycle engines, where the air enters at the circumference of the cylinder. The other methods, on the other hand, allow good filling and so permit of a high-power output, albeit at a small loss in thermal efficiency. For the maintenance of a flat volumetric efficiency curve allowing operation over a wide speed range, method (1) is the least desirable, and for this purpose use must be made of the other available means of creating air movement.

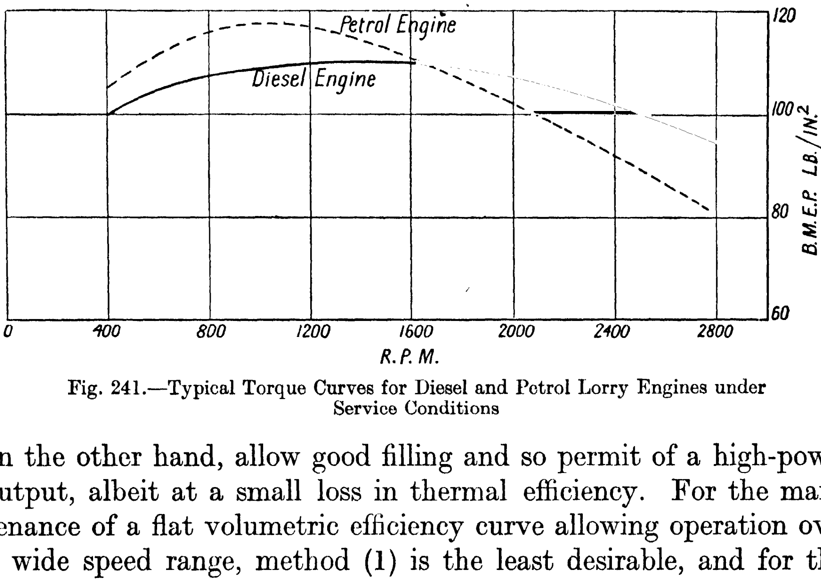


Fig. 241.—Typical Torque Curves for Diesel and Petrol Lorry Engines under Service Conditions

The modern high-speed Diesel engine under equivalent road service conditions can be designed to give a torque curve superior to that obtainable from a petrol engine of similar size and design for the upper half of the speed range, say above 1500 R.P.M.; and only slightly inferior at low speeds (see fig. 241).

Air Swirl.—The use of air swirl has been mentioned as one of the methods of giving good combustion during phase 3 of the combustion process, and the three ways of producing it have been touched upon in considering thermal and volumetric efficiency. Air swirl, however produced, has two main functions: (1) to assist in spreading fuel and partially-burnt products throughout the combustion chamber, (2) to feed fresh air constantly to the incoming fuel so that as much as possible of the air present may be usefully burnt. In other words, whereas air movement is required in a petrol engine to scatter the nuclei of flame and so spread combustion rapidly throughout the charge, in the Diesel engine the purpose of air movement is to bring the fuel and the air together. For rapid and complete combustion in the Diesel engine, the fuel must be swept away from the region of the fuel spray as it burns and unburnt air brought to the fresh fuel coming in. Unless this is provided for, the fuel entering later is blanketed and choked by burnt products, and only a relatively small quantity of air can be burnt.

Experience has shown that some degree of air movement is necessary whatever the system adopted, and that as the speed of operation is raised, the maintenance of good combustion with a high percentage of air burnt cannot be accomplished without employing a correspondingly high degree of air swirl. It is almost axiomatic that for very low-speed operation air swirl can be practically dispensed with; that for moderate speeds, say up to 1800 R.P.M. (the high-speed range of a few years ago), the most effective fuel dispersion of spray practically possible is inadequate, and a moderate degree of air swirl becomes necessary to assist in bringing fresh air to the fuel; and that for yet higher speeds, dispersion by fuel spray is totally inadequate and an intense air swirl must be used.

The reason lies in the fact that as the speed is raised the injection pressure to give the required depth of penetration within the time available is so high and the accompanying mechanical difficulties so acute as to put the use of the fuel jet alone entirely out of court as a means of bringing the fuel and air together. In these circumstances, as the fuel cannot be delivered to the air, the air must be brought to the fuel, and if the motion of the air is sufficiently rapid, it also assists in pulverization. A coarser fuel spray can then be used, which is all to the good from the point of view of penetration and nozzle life, and of uniformity of spray for long periods in operation.

In the writer's view Diesel engines are best classified according to the intensity of swirl and the type of fuel spray employed; in point of fact, when the type and number of fuel jets is known, the general behaviour and the characteristics of a given design are known also. Broadly speaking, when five or more holes are used in the injector, the required intensity of air swirl should be such that the ratio of the speed of rotation of the air to that of the crankshaft is 3 or 4 : 1; when two holes are used or two separate injectors having each a single hole, then an air swirl intensity of 5 or 6 : 1 will give the best results. Finally, when a single-hole injector is used the ratio must in normal circumstances be of the order of 9 or 10 : 1. These ratios are empirical values based on a particular method of measurement, and the values given are applicable only to the open type of combustion chamber using swirl.

There are indeed no high-speed Diesel engines in use to-day which do not in greater or less degree employ air movement generated either during the entry of air to the cylinder or during the compression stroke.

On the choice of method of swirl production depends the subsequent design. In the poppet-valve engine, swirl may be produced on the induction stroke by building a shroud on the underside of the valve head for some part of the circumference to prevent air passing into the cylinder in a direction contrary to that of the main stream. The valve port is placed at the side of the cylinder and the air is thus admitted tangentially. The inlet passage itself may be of such a form as to contribute to the general air direction required. The motion thus imparted to the air resolves itself into a special rotation about the cylinder axis of which the helix angle rapidly diminishes as the piston returns on the compression stroke. It is then usual to compress the rotating air into a combustion chamber of diameter somewhat smaller than that of the cylinder, thus augmenting the speed of rotation of the air.

In practice it is found that the maximum swirl rate attainable by this means without an impracticable sacrifice of power output by wire-drawing—due to loss of effective port area through the valve—is of the order of 5 or 6 : 1. In general, therefore, whenever induction swirl is employed in a poppet-valve engine a multi-hole fuel sprayer must be used. Even within the useful range of swirl intensity for this type, the reduction in breathing capacity and the consequent incongruity between volumetric efficiency and fuel supply characteristic combine to limit the useful range of speed. The use of the single sleeve-valve, on the other hand, lends itself admirably to the problem of overcoming the limitation in power output due to loss of valve area. The single sleeve, by virtue of its motion and the sideways opening of its ports, imparts a natural swirl to the entering air, which can be intensified or depressed at will by the use of deflectors placed in the inlet air belt. This is illustrated by fig. 242, which gives a horizontal section through the ports of an experimental sleeve-valve engine. A very wide range of swirl the figures are taken with a rotational swirl of 14,000 R.P.M., about eleven times the crankshaft speed.

Fig. 247 shows a similar graph of performance of another single-cylinder experimental engine of the same dimensions but designed for higher speeds, the normal crankshaft speed being 2200 R.P.M., corresponding to a piston speed of 2570 ft. per minute, at which speed the engine develops a maximum of 50 brake horse-power. Owing in part to the higher speed and in part to other factors, the mechanical efficiency of this engine is not so high as the former, and though the brake mean pressure and fuel consumption are not so good, namely, a maximum mean effective pressure of 108 lb. per square inch and a minimum consumption of 0·385 lb. per B.H.P.

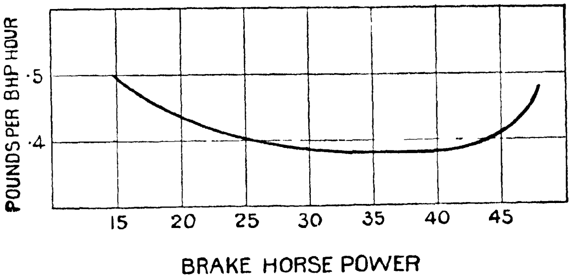


Fig. 247

r at 2200 R.P.M., yet the indicated figures for both engines are most exactly the same.

Figs. 248 and 249 show comparative indicator diagrams taken from the former engine at 1300 R.P.M. and from the latter at 2200 R.P.M., from which it will be seen that the influence of speed has little apparent effect on the indicator card. Figs. 250, 251 and 252 show photos and details of a moderately high-speed engine designed for commercial work and for Naval auxiliary services and manufactured in this country by Messrs. Peter Brotherhood Ltd. It is made with any number of cylinders from two to eight, and is intended to run at a normal speed of 900 R.P.M. The dimensions of the cylinders are 7½-in. bore, 12-in. stroke, and the power rating at maximum economy is 50 B.H.P. per cylinder, corresponding to an indicated mean pressure of 110 lb. per square inch. The fuel consumption at full load and normal speed and with a maximum pressure of 730–750 lb. per square inch ranges from 0·36 to 0·38 lb. per B.H.P. hour, depending on the number of cylinders; the larger the number, the higher the mechanical efficiency and the lower the fuel consumption. The normal piston speed is 1800 ft. per minute, but several engines are in regular service running at 1000 R.P.M., corresponding

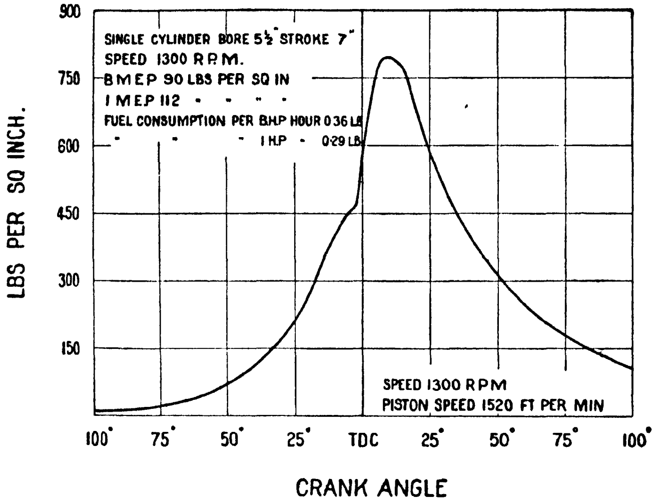


Fig. 248

to a piston speed of 2000 ft. The sectional drawings (figs. 253 and 254) clearly show all the mechanical details. The single sleeve is operated by means of an overhanging beam driven by a lay-shaft

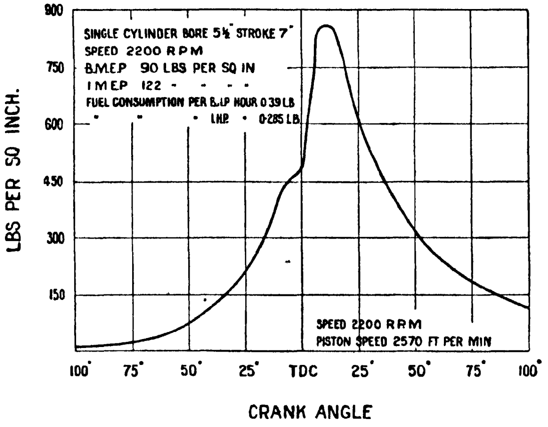


Fig. 249

running at half engine speed. This gives the sleeve an elliptical motion with the major axis vertical. By employing an elliptical motion of this nature it becomes possible to use a considerable number of relatively high and narrow inlet ports—actually five in number—a form and arrangement of ports which is best suited to provide a tangential entry for the air.

The crankshaft is of the built-up type, with case-hardened pins and journals. The webs are shrunk on to the journals, but the pins are detachable and held in place by means of clamp bolts. This allows of the use of unsplit floating bearings for the connecting-rods, and also renders dismantling of the crank quite easy, for any crankpin can be withdrawn and replaced through the inspection doors in a comparatively short time merely by removing the adjacent main bearing cap and slackening off the two clamping bolts.

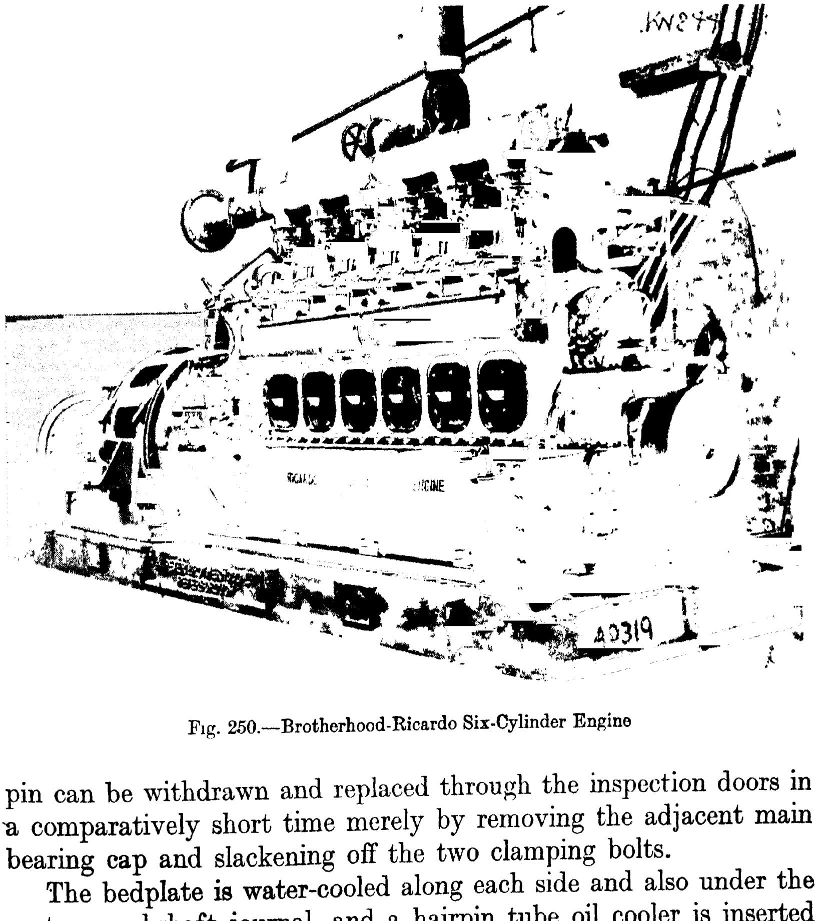


Fig. 250.—Brotherhood-Ricardo Six-Cylinder Engine

The bedplate is water-cooled along each side and also under the centre crankshaft journal, and a hairpin tube oil cooler is inserted the full length of one of the bedplate water jackets. One end of the bedplate is divided off into a separate compartment to form the main oil tank containing some fifteen gallons of lubricating oil. In this compartment are fitted two rotary oil pumps, one of which draws oil from the crank pits and delivers it through an Auto-Klean filter into the main tank; the second pump draws from the tank and delivers the oil under a pressure of about 12 to 15 lb. per square inch through the oil cooler and thence to all the important bearings.

Compression Swirl Systems.—There are to-day very large numbers of engines in service which use the second method of swirl production, namely, air swirl produced on the compression stroke by means of the compression of air into a combustion chamber which is designed to accommodate an orderly air-flow and connected with the cylinder by a short tangential passage as illustrated in figs. 255 and 256. This system, a development of a very old idea, has been produced in many forms and has features which give it a wide sphere of usefulness. From a mechanical standpoint it has the advantage that it can be applied to any size of cylinder down to one of very small capacity. Further, as will be readily seen from the drawing, the arrangement allows plenty of room for the valves, and as there is no question of making use of the valves or valve parts to create the swirl, no restriction need be placed on the breathing capacity of the engine. In the earlier forms there was some difficulty in

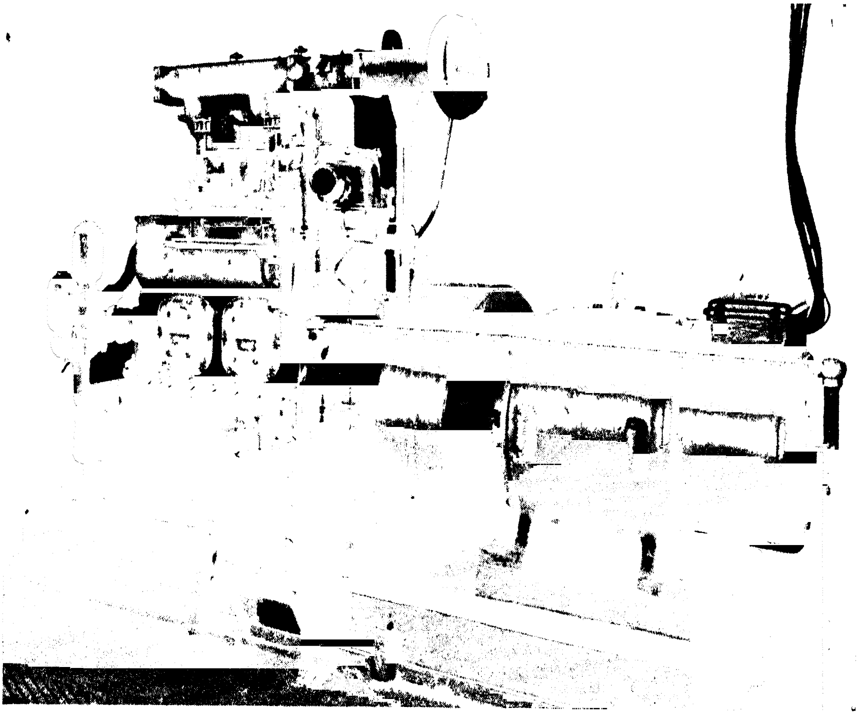


Fig. 251.—R 100-2 Engine and Dynamo on Combination Bedplate

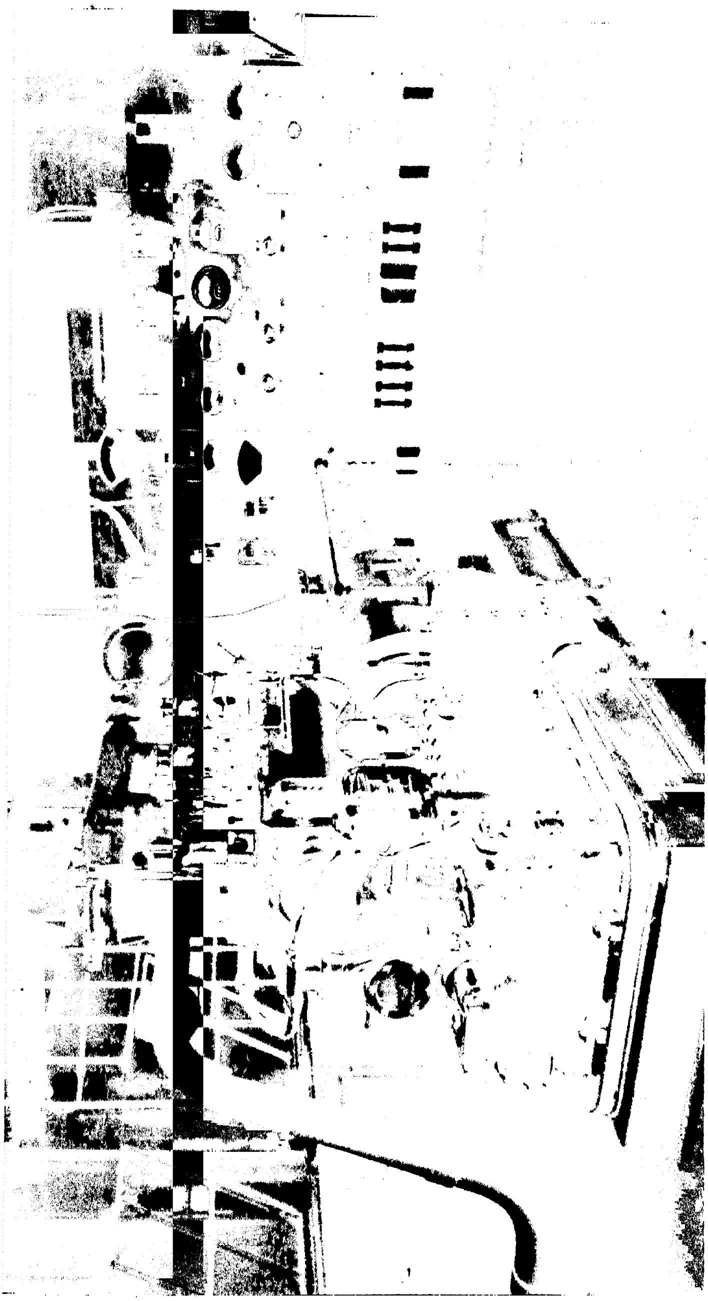


Fig. 252.—R 101-2 Engine Dynamo Unit and Power Station Switchboard

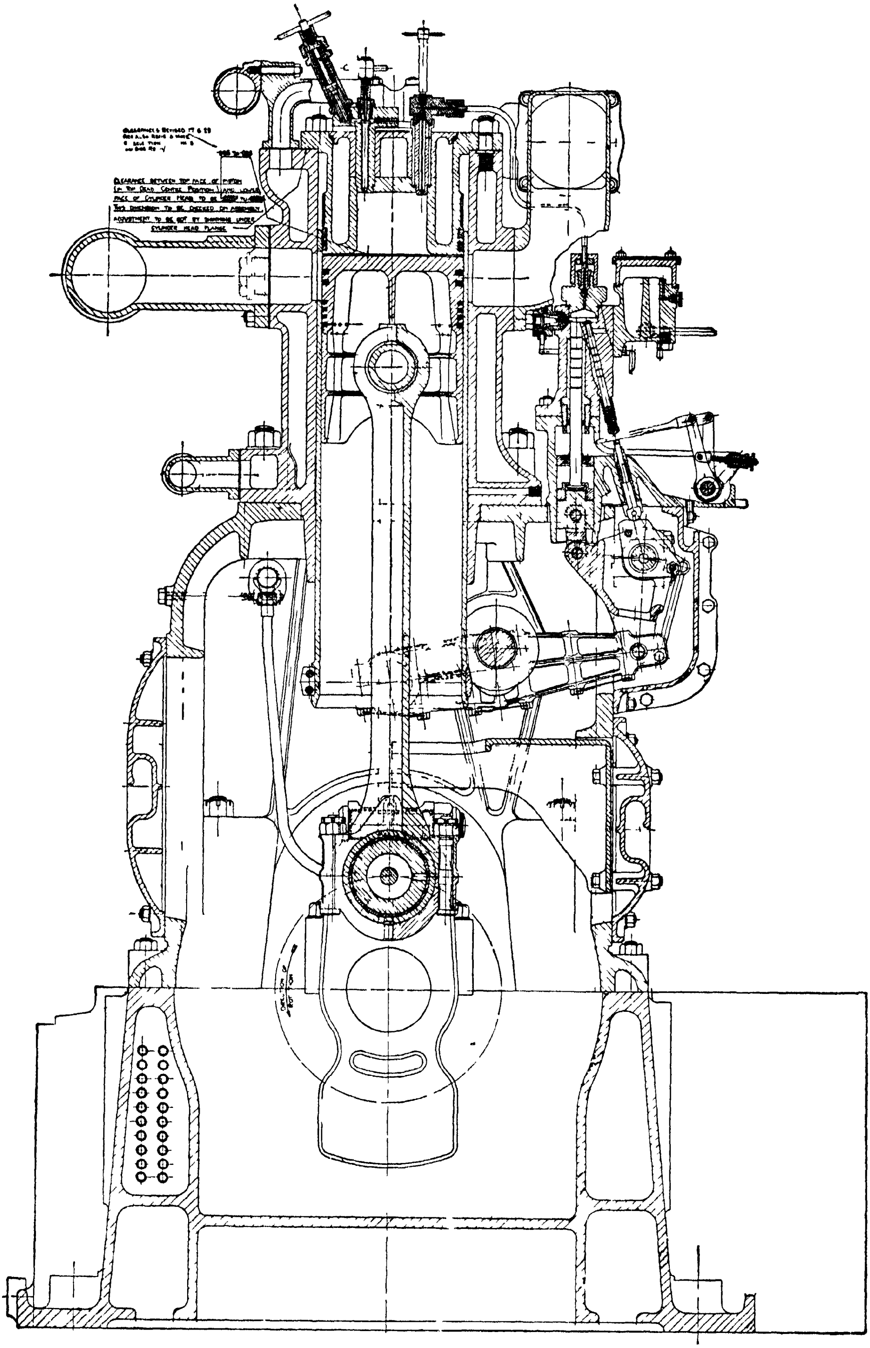

Clearance between top face of piston (in top dead centre position) and lower face of cylinder head to be
This dimension to be checked on assembly.
Adjustment to be got by shimming under cylinder head flange
Direction of rot

Fig 253 —Arrangement of $7\frac{1}{2}' \times 12''$ Brotherhood Ricardo High Speed Diesel Engine Cross Section

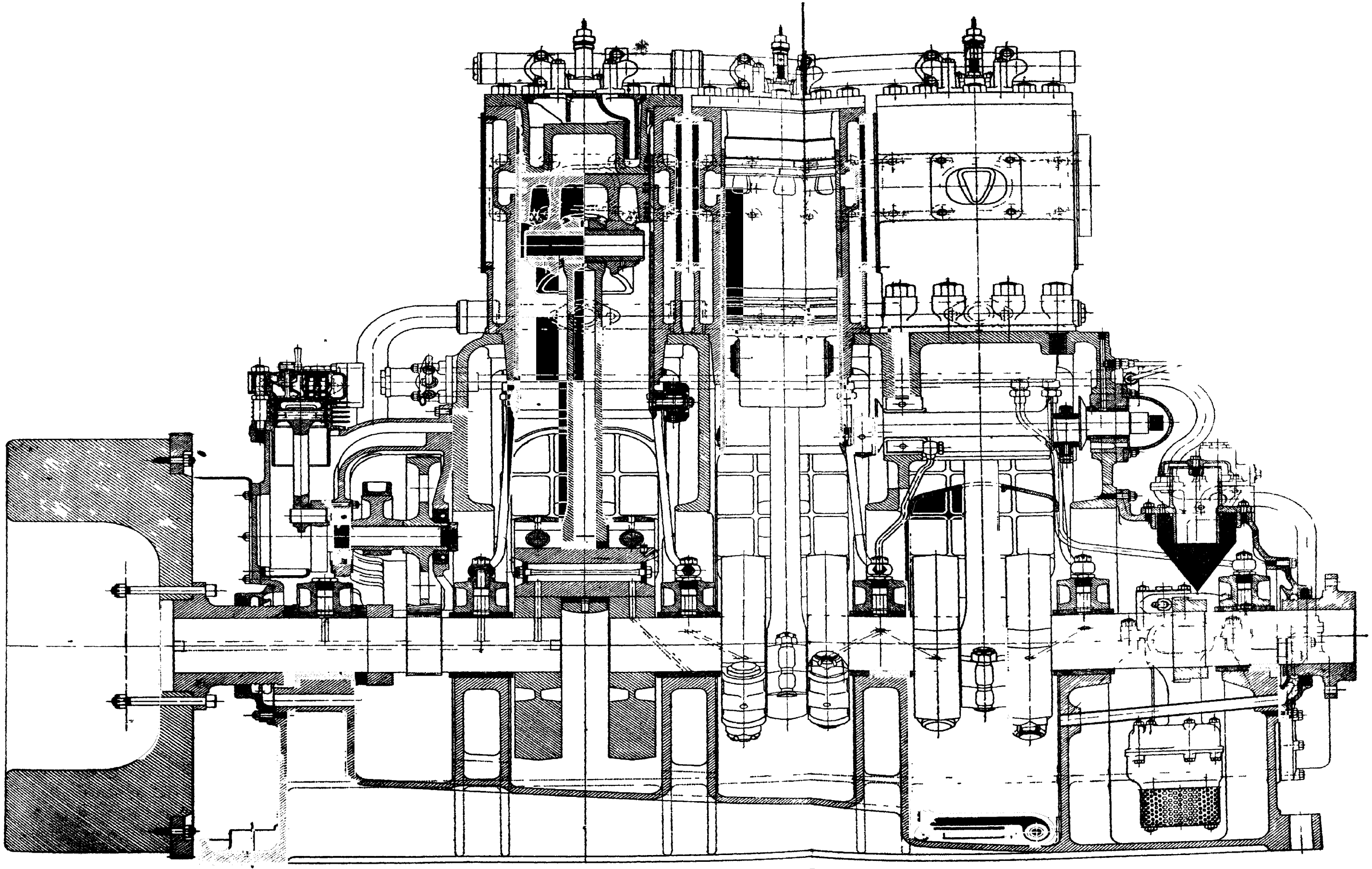
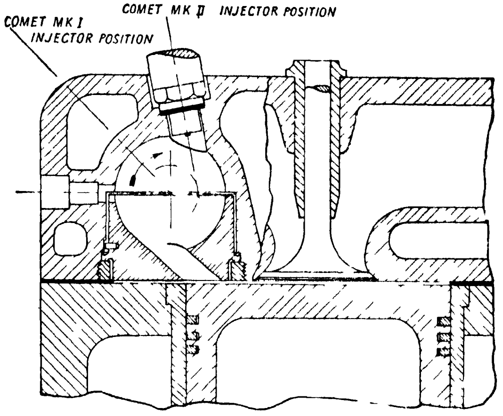


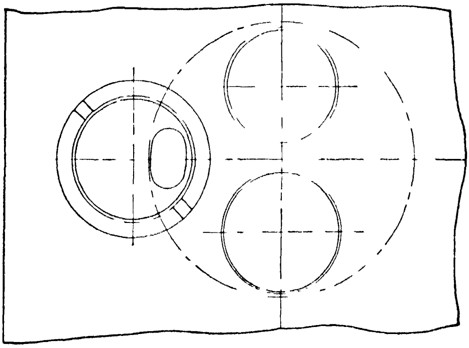
Fig. 254.—Longitudinal Arrangement of 7½" × 12" "Brotherhood-Ricardo" High Speed Diesel Engine

bringing all the air into contact with the fuel, and hence the power output was somewhat limited. As now developed and used, however, a very high proportion of the air is burnt and the system gives the highest known power output from a given cylinder capacity.

The illustrations show pictorially landmarks in this development. As at first designed the form of chamber was substantially as shown in fig. 255, but the injector was in such a position that the centre line of the fuel spray passed through the centre of the sphere. The broken line shows this early position. In this form, known as " Comet Mk. I ", a moderate power output was obtained, but a characteristic of the system was the early arrival of carbon in the exhaust as the fuel quantity was increased from light load



SECTIONAL ELEVATION



INVERTED PLAN OF HEAD SHEWING RELATIVE POSITIONS OF HOTPLUG AND VALVES

Fig. 255.—Comet Mk. I and II Cylinder Heads

operation. Subsequently it was found that a greater proportion of the air could be utilized and a higher load carried with clean exhaust by placing the fuel in such a position that the early products of combustion carried round by the swirl did not blanket the later supplies of fuel. The fuel jet was accordingly aimed at a point as far as possible circumferentially to windward of the injector, while fulfilling the two conditions of (1) spraying the fuel on to the hot surface and (2) avoiding excessive deposition on the wall from the fringe of the jet. In this way the greatest possible amount of clean air, uncontaminated with burnt products, could be fed past the incoming fuel. Fig. 255 shows the arrangement with the fuel jet aimed in this "downstream" direction, called the "Comet Mk. II" cylinder head.

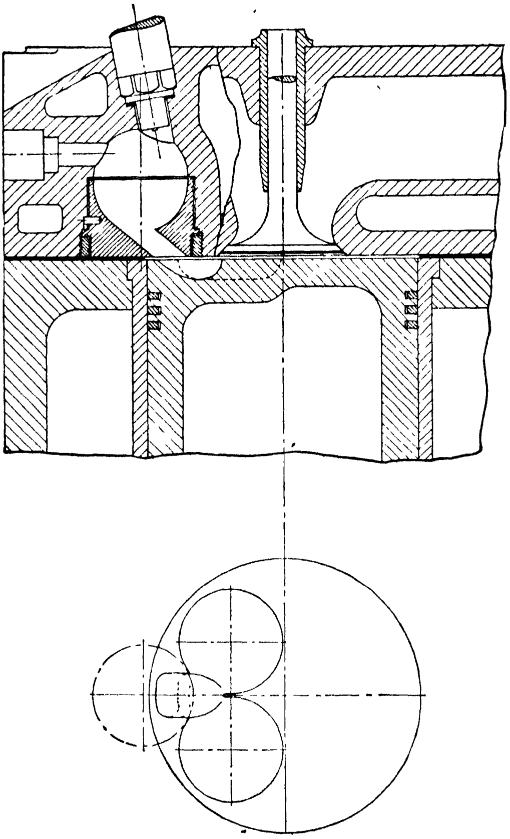


Fig. 256.—Comet Mk. III Combustion Chamber

Fig. 256 illustrates the next step in the development. As was mentioned earlier, the compression of air into the combustion chamber and the subsequent egress of the burning products at high velocity from the throat involves a loss in thermal efficiency, and by reducing the proportion of the charge to undergo this process to 50 per cent, as compared with 75 to 80 per cent in the case of the " Comet Mk. II ", a further improvement was made. The displaced volume was accommodated in a cavity in the piston. The cavity, after much experiment, was made in the form of two circular shallow dishes partially divided by a promontory to separate the stream of burning gases issuing from the throat into smaller streams entering the dishes tangentially. The air contained in the piston cavity was in this way set into a swirl and at the same time well supplied with partially burnt fuel seeking complete combustion. Tests showed that as much as 90 per cent of the available air could be burnt by this arrangement at the clean exhaust limit. One or two cases gave unusually good results, up to 95 per cent. This system marked a big advance in the attainment of high-power output with good thermal efficiency, a combination which had proved difficult of attainment. With this form of combustion chamber indicated mean pressures of over 160 lb. per square inch have been obtained without supercharge.

The " Whirlpool " combustion chamber (fig. 257) was evolved from the accumulated experience gained in the development of the " Comet " system, with an eye to the provision of easier starting and smoother running without loss in performance.

When the second " Comet " chamber was tested, it was noted that with the improvement in power output the engine had lost some of the good starting and smooth running qualities it possessed when the fuel jet had been directed towards the centre. This was found to be due to the position in the chamber of the fuel to be first injected. In the first arrangement—" Comet Mk. I "—some fuel reaches the inner end of the passage leading into the sphere, the point of highest temperature in the chamber at the end of the compression stroke. Moreover, owing to the low piston velocity immediately previous to T.D.C., the velocity of gas through the throat is rapidly falling. These two conditions combine to provide a zone which is favourable to a reduction in delay period and easy starting. Unfortunately, in the " Comet Mk. I " cylinder head the advantage obtained in starting by supplying fuel to this point was paid for by a curtailment of power output and efficiency, because the burnt products from the fuel thus early ignited were at once carried round to the region of the injector to smother the incoming fuel.

These observations, and the theory built upon them, gave rise to the "Whirlpool" combustion system, in which two holes are used. One hole fed by the fuel jet provides a good starting zone and eliminates the Diesel knock, which is usually severe with a cold engine, while the other hole serves to provide a rearguard of uncontaminated air to complete and clean up the combustion.

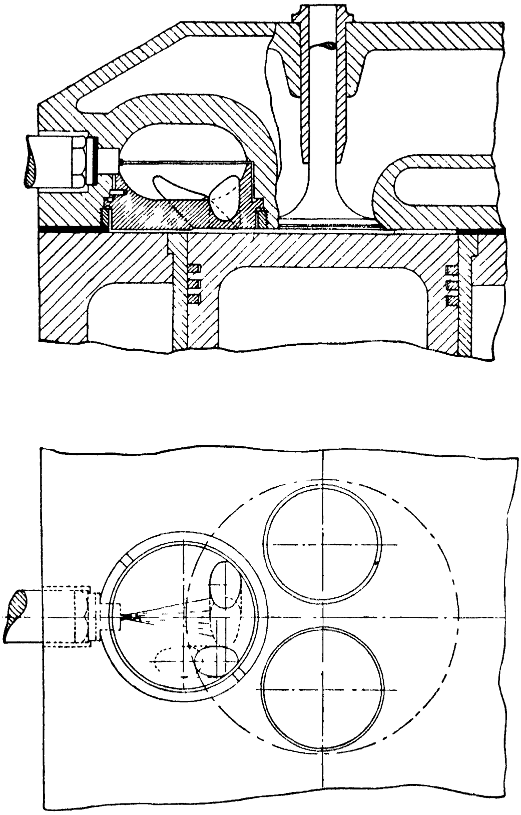


Fig. 257.—Whirlpool Combustion Chamber

Owing to the high degree of swirl, of the order of 25–30 times the crankshaft speed, in the combustion chamber, pulverization of the fuel is relatively unimportant and a coarse and low-pressure fuel spray may be used.

The lower half of the combustion chamber shown as a separate piece in the designs illustrated in figs. 255, 256 and 257 takes the form of a loose member of heat-resisting steel, insulated from the water-cooled head by a small air-gap. It not only acts as a heat regenerator giving combustion control, but also has the valuable function of eliminating smell from the exhaust by breaking down the aldehydes formed as products of partial combustion. Use is made of its heat capacity to raise the compression temperature and so reduce the delay period in the manner mentioned earlier. This, by its influence on the delay period, enables the engine to operate with fixed injection timing, with moderate rate of pressure rise (in the case of the "Whirlpool" head a very low rate) and with a fixed peak pressure throughout the whole range of load and speed. With this form of control combustion efficiency is maintained up to speeds of 4000 R.P.M. and over. From figs. 230 and 231 it will be seen that at all speeds and loads the temperature is well above 300°–400° C. By maintaining a combustion chamber surface temperature above this value the stabilization of the products of partial combustion due to the flame impinging on a relatively cold surface (below, say, 300° C.) is entirely prevented. These products, if allowed to pass out in the exhaust, give a pungent and acrid smell.

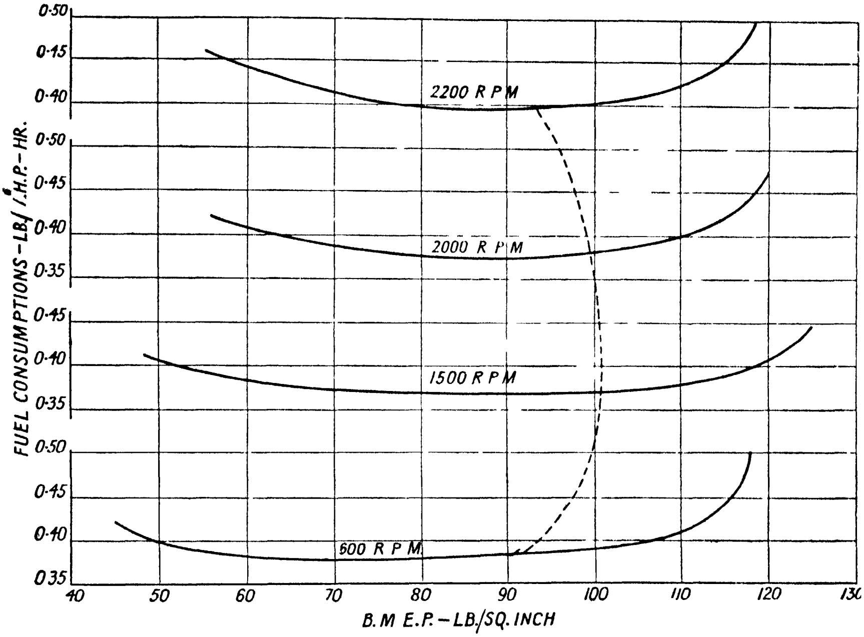


Fig. 258.—Fuel Consumption-Load Curves for Comet Mk. III Road Vehicle or Marine Compression-ignition Engine. Bore 4¾ in., Stroke 5½ in.

Fig. 258 gives fuel consumption curves obtained over the load nge at various speeds from 600 to 2200 R.P.M. for an engine fitted with the "Comet Mk. III" system. The broken line is drawn through points on the curve corresponding with the torque curve of an engine running with the fuel pump control fixed to give 100 lb. per square inch B.M.E.P. at 1500 R.P.M. This position of the control represents a full-load road setting. Fig. 259 gives similar curves

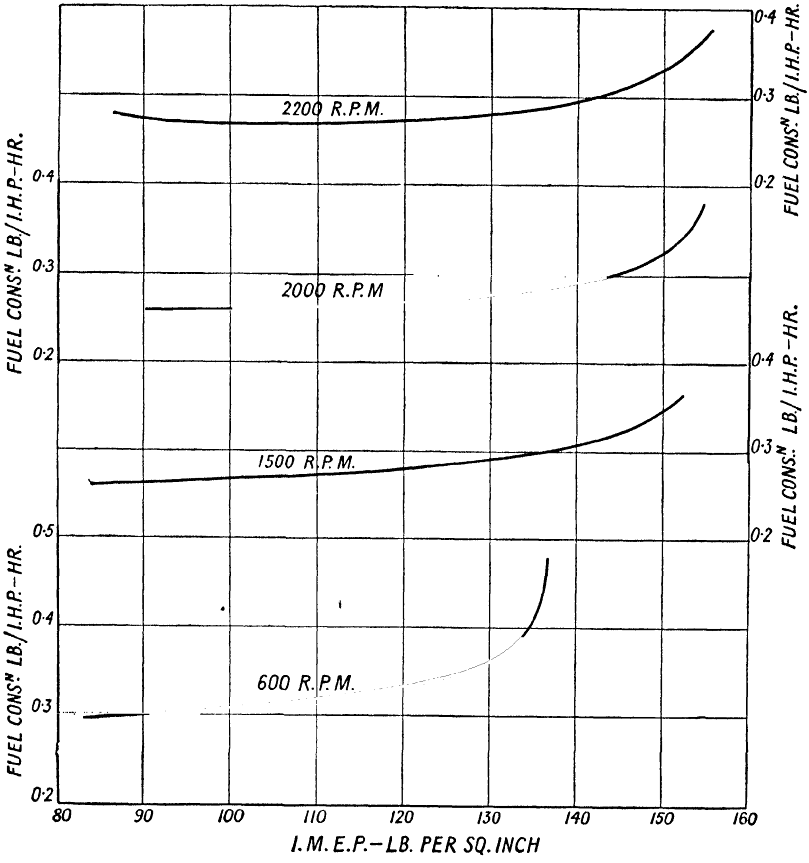


Fig. 259.—Indicated Consumption Curves for Engine fitted with Comet Mk. III Combustion Chamber

plotted on an indicated basis, and fig. 260 gives the indicated thermal efficiency over the speed range for the three load conditions: (*a*) at the limit of clean exhaust, (*b*) with pump control fixed at full rated load, and (*c*) with fuel supply for ¾ full-load. The engine was designed to operate at a normal speed of 1500 R.P.M. up to a maximum of 2200 R.P.M. The curves show that throughout this range the Indicated Thermal Efficiency continues to rise with increase in speed.

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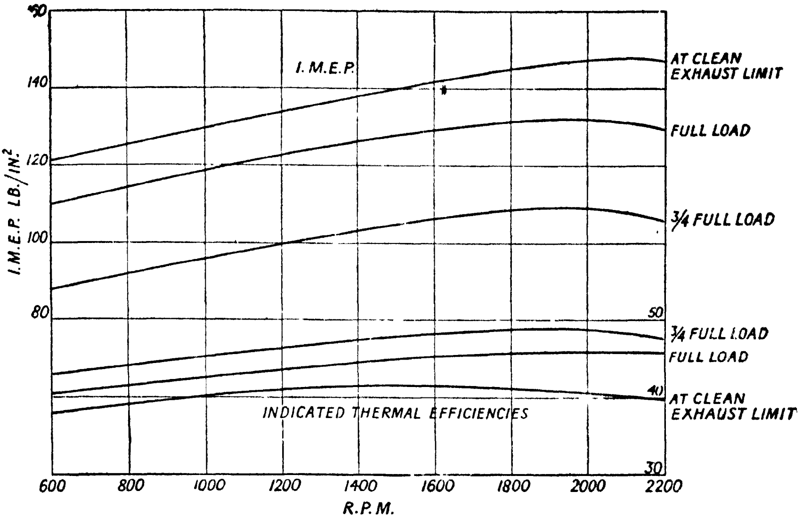


Fig. 260.—Curves of Indicated Mean Pressure and Thermal Efficiency for High-speed Diesel Engine fitted with Comet Mk. III Combustion System

Indicator Diagrams with various Types of Compression Swirl Chambers using an insulated Hot Plug Combustion Control.—Typical diagrams are given in fig. 261 for the three "Comet" systems already described. The diagrams were taken by

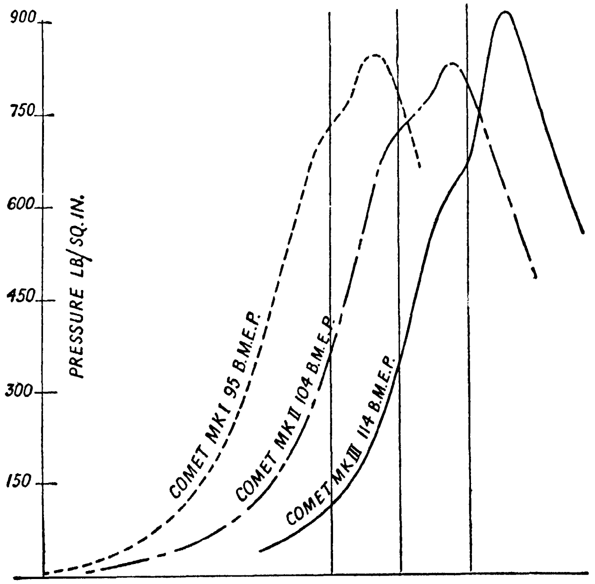


Fig. 261.—Indicator Diagrams of Comet Mks. I, II, and III Combustion Chambers. 1500 R.P.M.

a Farnborough Electrical Indicator with the pressure-recording unit screwed into the wall of the spherical chamber. In each case the load was adjusted at the limit of clean exhaust at a speed of 1500 R.P.M., piston speed 1375 ft. per minute.

To illustrate the effect of using two throat holes in the combustion chamber, some indicator diagrams taken on the "Whirlpool" cylinder head are given (fig. 262). The transition from the compression line to the pressure line of phase 2 is a smooth curve in every case, whether the engine is operating at the comparatively low load of 45 lb. per square inch brake mean pressure or at the lower end of the speed range. Indicator diagrams obtained by injecting the fuel very late are included to show the influence o;

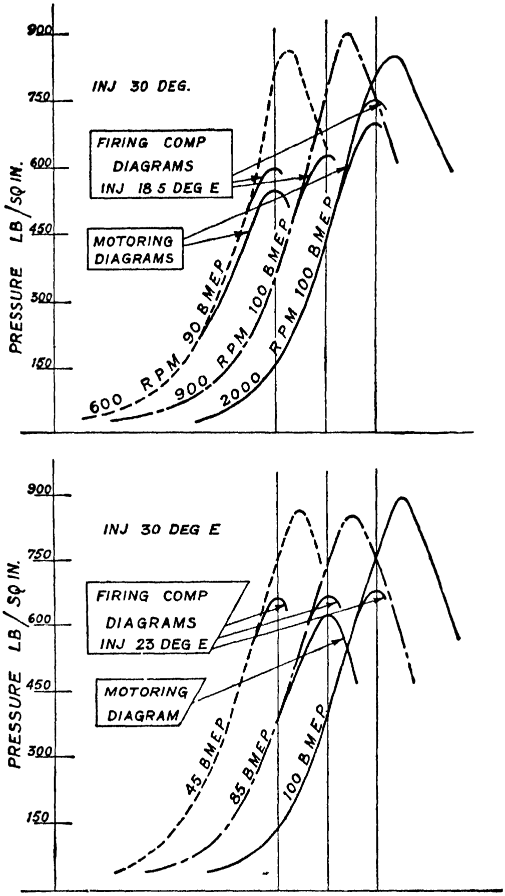


Fig. 262.—Indicator Cards over Speed and Load Range. Whirlpool Combustion System

the insulated hot plug on the compression pressure under "firing" conditions.

The diagrams which were taken with fixed injection timing over the load range and over the speed range also show the small variation in maximum cylinder pressure over a wide change in operating conditions.

Progress of the High-speed Diesel Engine.—It was in the heavier class of vehicle that the high-speed Diesel engine made its début on the road, and it is mainly in this field that the change-over from petrol to fuel oil has taken place. Upwards of 80 per cent of the total number of heavy passenger vehicles put into service in this country are now fitted with Diesel engines On the other hand, the smaller classes of vehicle are practically untouched.

The development of the small high-speed cylinder suitable for use in light vehicles is quite recent, and a more widespread application of this type awaits the gradual growth of confidence which is the necessary precursor of acceptance on a large scale.

It was a natural evolution from the days of the large slow-moving Diesel engines that the first high-speed applications should be of relatively large cylinder dimensions. Of moderate speed and output, these newcomers colonized the field most suited to them. Subsequently it was found that by making more use of air movement small cylinders could be used and very much higher speeds attained. The mechanical difficulty of finding room both for an injector over the centre of the cylinder and two overhead valves practically rules out the open type chamber in the very small sizes; either the pre-combustion system or some form of compression swirl chamber should be used.

The outstanding British development in these early days was the Gardner engine, which, thanks to superb workmanship and attention to detail, very early became a marked success as a power unit for vehicles of the heavier class and has so continued to the present day. At the same time the single sleeve-valve high-speed industrial engine illustrated in figs. 253 and 254 was developed. It has given good service in marine, auxiliary and small stationary power plants.

The lines on which subsequent development proceeded have been indicated and many engines of various types for road vehicles were produced.

A typical modern design is shown in fig. 263 (pp. 424, 425). The cross-section of this engine was shown in fig. 74. Developed for heavy commercial vehicles, this six-cylinder engine, bore 4¾ in.,

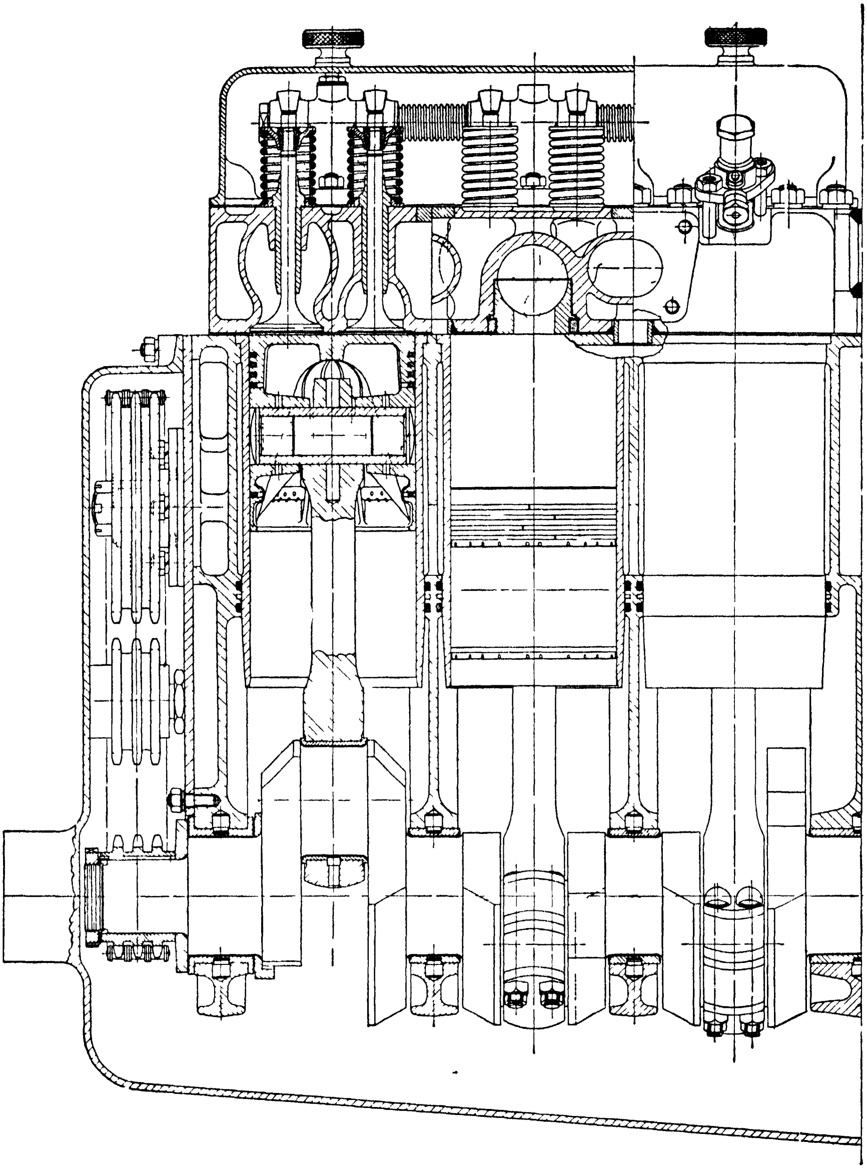
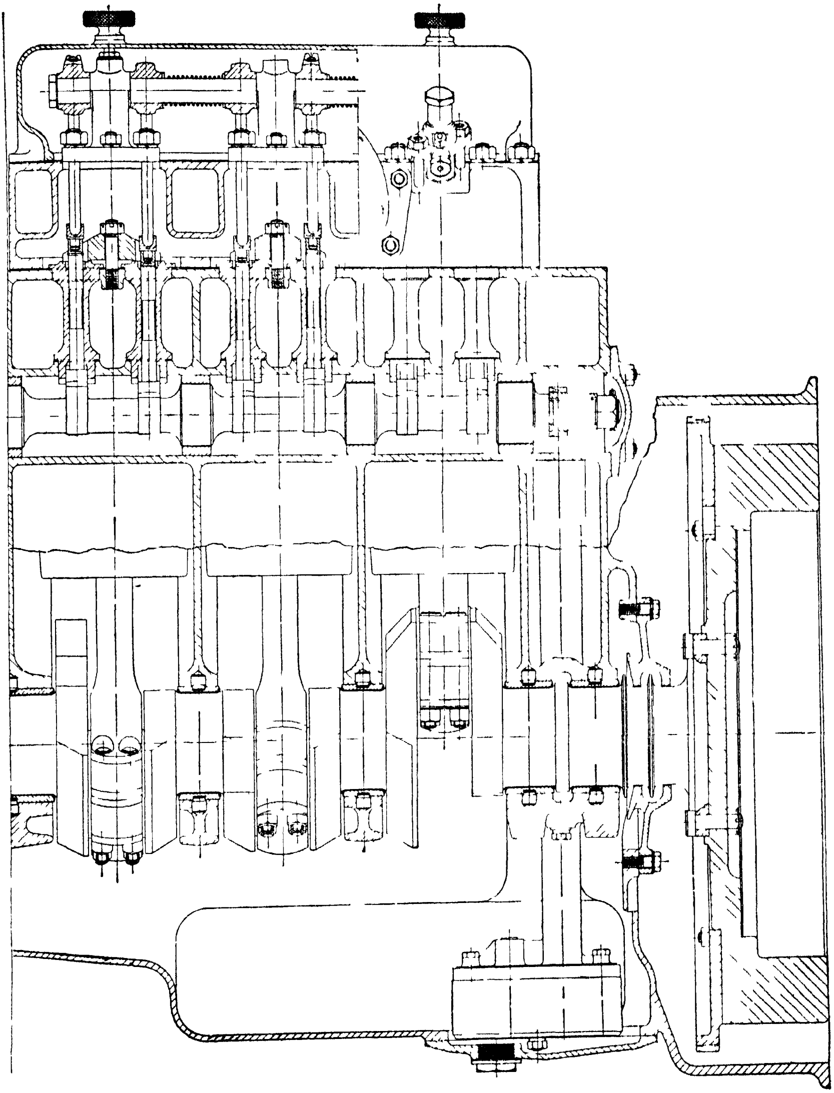


Fig. 263.—Longitudinal arrangement of six-cylinder Compression-ignition Engine, $4\frac{3}{4}$-in. bore, $5\frac{1}{2}$-in. stroke

stroke $5\frac{1}{2}$ in., which weighs 1500 lb., has a power output of 170 H.P. at 2100 R.P.M. Performance curves are given in figs. 264 and 265. The combustion head is of the type in which 50 per cent of the clearance space is contained in the spherical chamber and throat, while the remaining 50 per cent occupies the cavities in the piston and the head clearance. The injector is located almost vertically and injects downstream with regard to the air movement. The fuel nozzle is of the pintle type.

The "Comet" type of combustion chamber lends itself well to incorporation in very small cylinders, as is shown by the small



vehicle engine illustrated in figs. 266–269 (pp. 427–429). The author is indebted to S.A. André Citroen for the views and sections of this engine given in the figures. Fig. 266 clearly shows the form of combustion chamber, which in this case is of the type known as

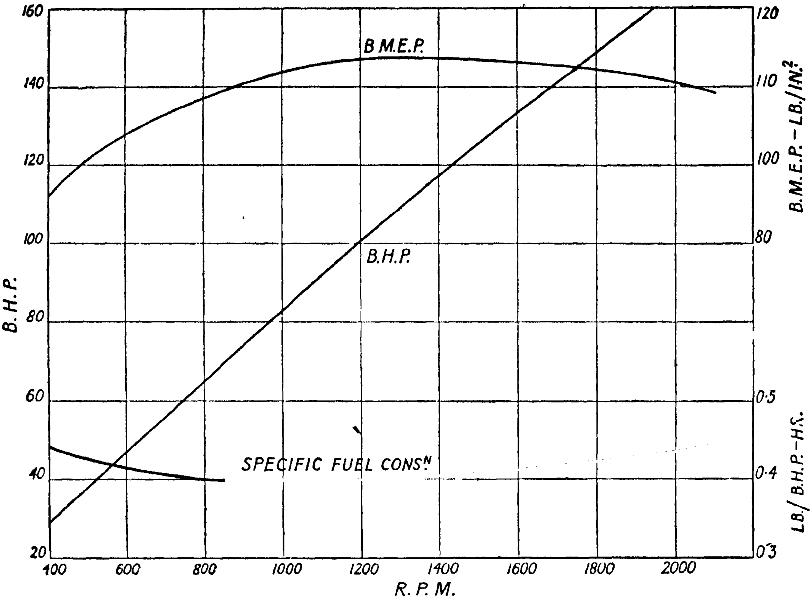


Fig. 264.—Power Curve of Six-cylinder Compression-ignition Engine ($4\frac{3}{4}$ in. × $5\frac{1}{2}$ in.) fitted with Comet Mk. III Combustion Chamber

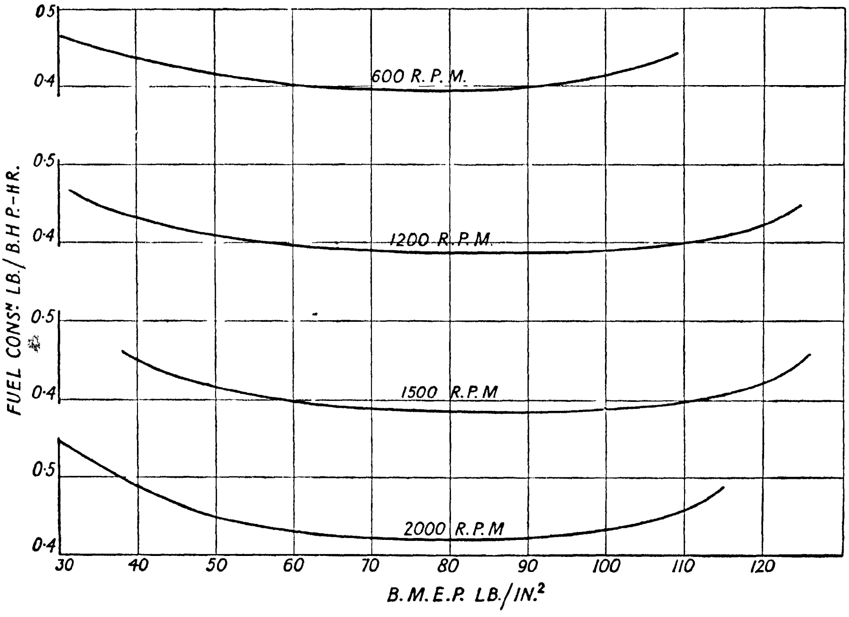


Fig. 265.—Fuel Consumption-Load Curves at various Speeds on Six-cylinder Engine ($4\frac{3}{4}$ in. × $5\frac{1}{2}$ in.) fitted with Comet Mk. III Combustion Chamber

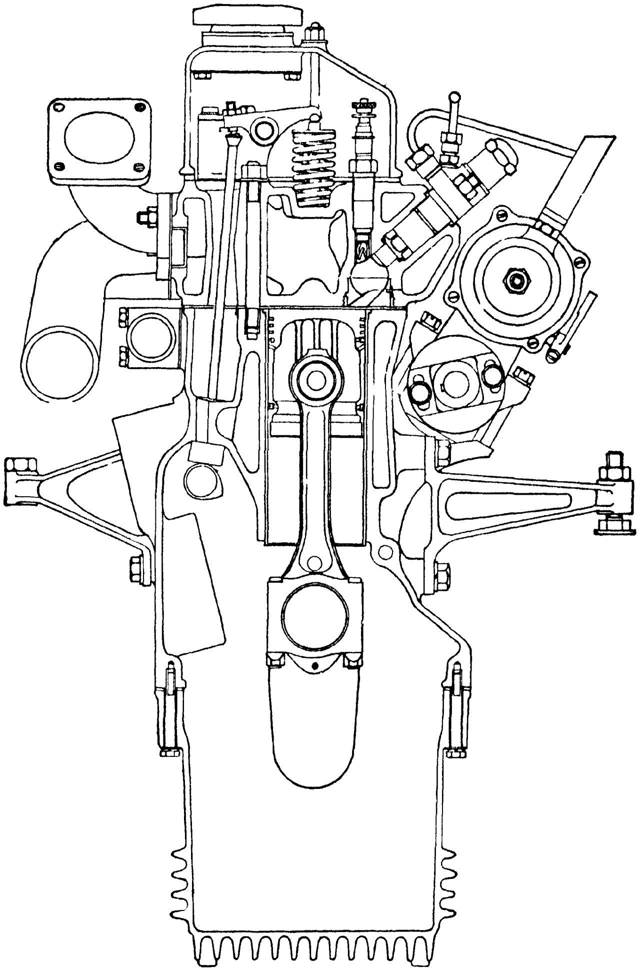


Fig. 266.—Citroen 1·7 litre Diesel Engine for Light Vehicles—cross section

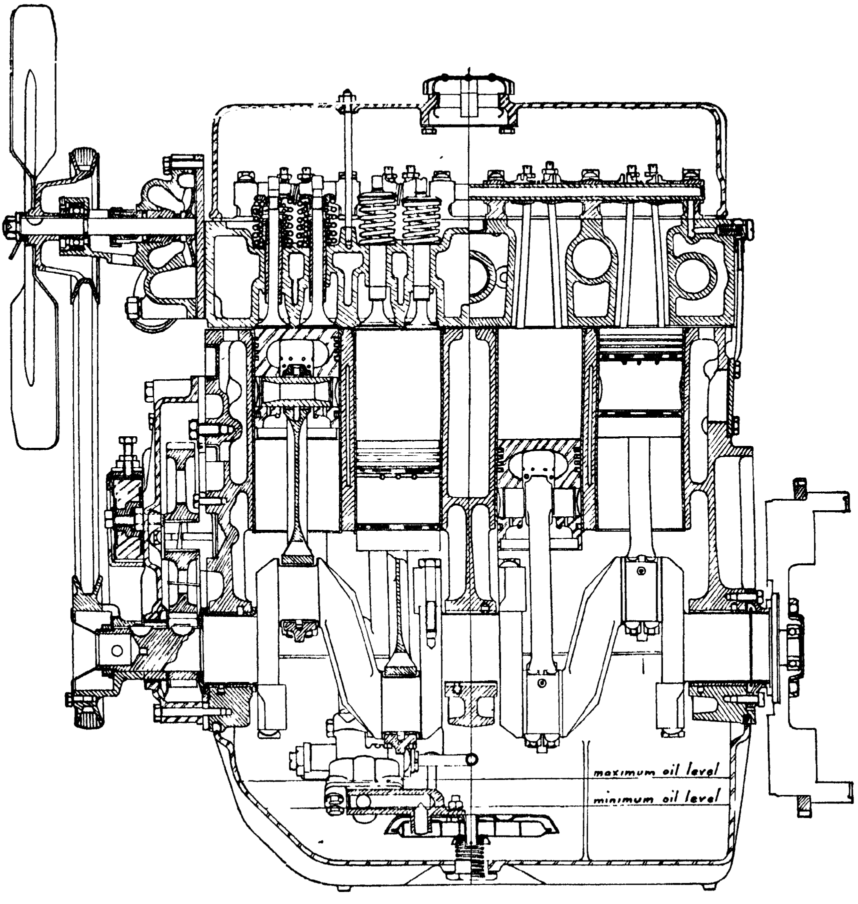


Fig. 267.—Citroen 1·7 litre Diesel Engine for Light Vehicles—longitudinal section

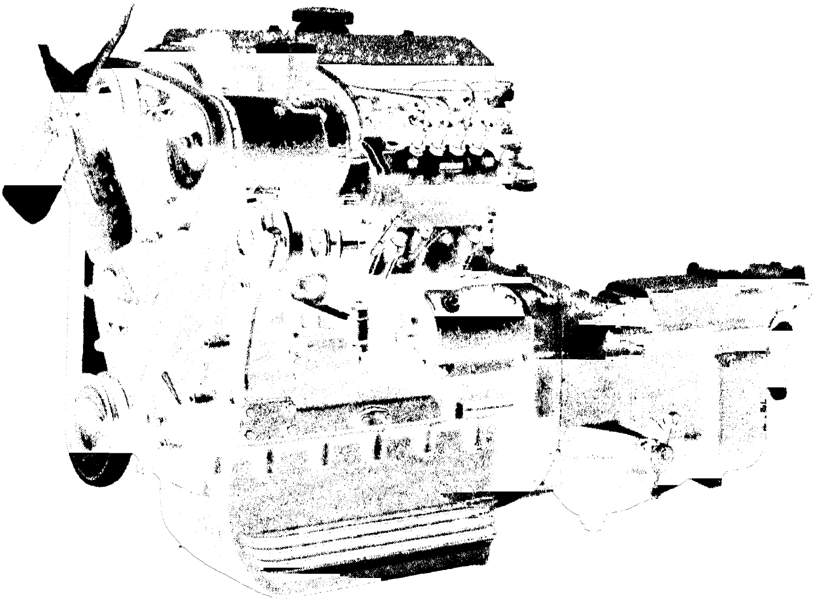


Fig 268

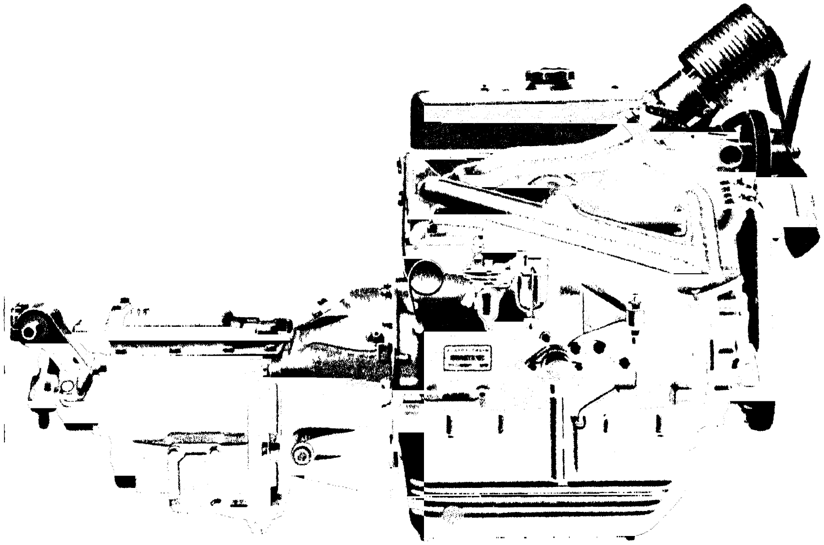


Fig. 269.—Citroen 1·7 litre Diesel Engine for Light Vehicles

"Comet Mk. II". The performance of the single-cylinder test unit on which the engine was based is given in fig. 270. We see that the indicated thermal efficiency continues at its maximum value beyond a speed of 4000 R.P.M., showing that combustion efficiency is maintained up to a very high speed.

The subject of high-speed Diesel engines has attained the dignity of a separate branch of mechanical engineering, and a vast amount of research data and practical experience has been accumulated. In these circumstances it has not been possible to attempt in one chapter more than a brief outline of the general principles which have been established in the course of the development of this latest class of high-speed engine.

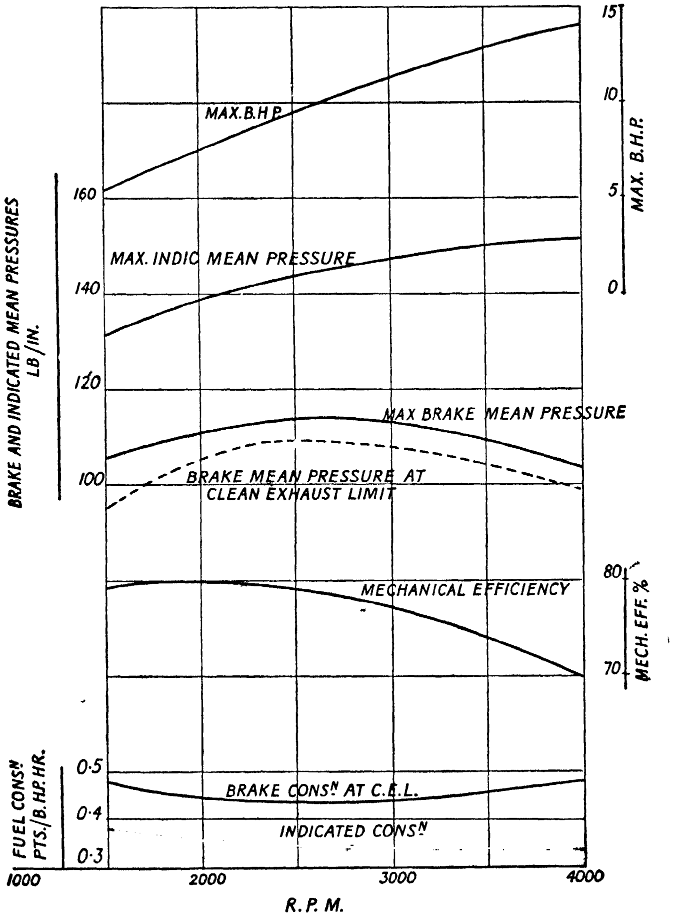


Fig. 270.—Tests on small (75 mm.) Single-cylinder Compression-swirl Experimental Engine. C.R. 19·5 : 1. Maximum Cylinder Pressures allowed to rise to optimum, viz. 930–990 lb./sq. in.

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